CERTIFICATION OF A HOT TAPPING DRILLING DEVICE

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Bachelor's thesis
May 2013
Environmental Engineering

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ABSTRACT

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Purpose and aim of this bachelor thesis has been to find a guideline for the obtainment of the EC Type-Examination approval for the product TONISCO Jr. supplied by the company TONISCO System Oy. TONISCO Jr. is the name of a drilling device for pipes under pressure in a range from DN20 to DN100. A certificate can be given by the VdTÜV which is one of the notified bodies for different kinds of pressure equipment. After a successful pressure test, a product is allowed to be marked with a CE-Mark and a test certificate is handed out to the applicant company. The CE-Marks and the test certificates purpose is to get an independent assessment for a product by a different organisation. They are often required by customers or service societies in work environments that involve risks of damages or accidents.

In order to get an approval from the VdTÜV, certain requirements have to be fulfilled, depending on the purpose and the field of use. In this bachelor thesis, the approval for district heating has the priority and is examined.

After an explanation about the general structure of Hot Tapping, the required documents for the EC Type-Examination of the drilling device are mentioned. Necessary documents as for example a translated manual or the categorization of the drilling device according to the PED had been part of the thesis work. Furthermore, a calculation of the pressure parts is done, whereby they will be submitted to TONISCO System Oy to be verified and possibly used for the TÜV application.

In chapter 6, contact areas that can hardly be calculated by traditional hand calculations are examined and the effects of notches are analysed by the FEM application ANSYS® Workbench.

Key words: Pressure Equipment Directive, EC-Type Examination, Armatur 100, VdTÜV, FEM, Strengths Calculations, ANSYS®

CONTENTS

ABBREVIATIONS

PED	Pressure Equipment Directive
VdTÜV	Verband der Technischen Überwachungs-Vereine
DIN	Deutsches Institut für Normung
EN	Europäische Norm (European Standard)
FEM	Finite Element Method
p_v	Vapour pressure of water at a defined temperature
δ	Thickness of a slice element
σ_r	Stress in radial direction
σ_t	Stress in tangential direction
σ_l	Longitudinal stress
σ_v	Equivalent Stress
p_i	Inner fluid pressure
σ_{hb}	Hole bearing stress
τ_{shear}	Shear stress
$f_{allowable}$	Maximum allowable stress (including security factor)
ν	Transversal contraction
C_0	Maximum static force for the bearing
p_{flank}	Flank pressure of the thread
W_b	Section modulus
r	Radius
r_i	Inner radius of a pipe
r_a	Outer radius of a pipe
α_1	Hole bearing factor
α_{bp}	Plastic deformation factor for a part without notch
n_{pl}	Plastic support factor
σ_b	Normal stress caused by bending
f_σ	Conversion factor according to FKM-directive
α_k	Notch factor
R_{pmax}	Maximum yield strength (for steel $R_{pmax} = 1050N/mm^2$)
R_p	Maximum allowable stress
$R_{p0,2t}$	Yield strength at operating temperature
P	thread pitch

1 INTRODUCTION

Pressure equipment of every kind involves risk, as the pressure and temperature of a fluid stresses a part or lowers the resistance of the supporting material. In order to minimize the dangers for the user, it is reasonable to let the documents and drawings be controlled and the product be tested by an impartial, independent notified body. This authority needs to follow and approve the content and thus, all documents have to be supplied according to their guidelines. Regarding pressure equipment, most of the used regulations are based on a European administrative fiat, the Pressure Equipment Directive. In Germany, the VdTÜV is one of these notified bodies which is allowed to test pressure equipment and to hand out certificates for EC Type-Examinations and other approvals. After a successful pressure test, the pressure equipment can be marked with a CE mark.

Dimensioning and design of pressurized parts have to be made according to the PED annex I and all the listed requirements need to be taken into account. The dimensioning is allowed to be made by three different methods, whereby the content and the procedure of these methods are more or less in the responsibility of the authorized person(s)of the company.

According to the PED, the use of formulas or analyzing methods is permissible, but their correctness needs to be assumed by the responsible person(s). Generally, basic hand calculation methods are the usual way of dimensioning and design, but since the use of FEM analysis programs, such as ANSYS® Workbench, are nowadays very typical and the exact stress at many locations is not easy to be determined in a conservative way, it is very reasonable to use them.

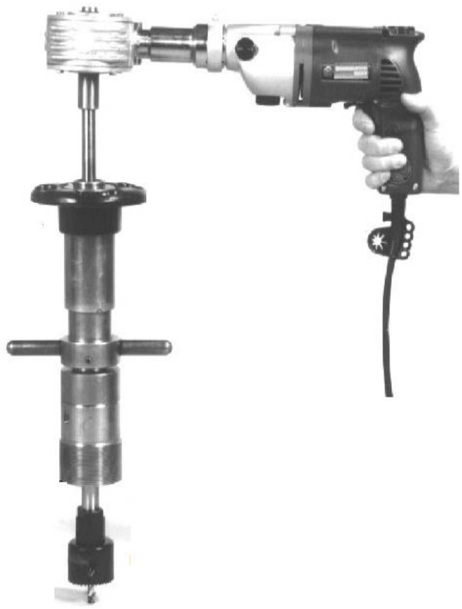
2 TONISCO System Oy

The 1969 established company TONISCO System OY is mainly developing, selling and delivering pipe line maintenance tools, whereby a maintenance service is also offered. As their main innovation, they are supplying drilling devices for Hot Tapping Systems which are produced by subcontractors, but also presses for line stopping or valves are part of their products. TONISCO System Oy is a family company located in Tampere (Finland) from where drilling devices in a range from DN15 – DN700 are supplied to costumers all around Europe.

During the first 10 years, the company was based on the domestic market until they decided to offer their products to the European market as well. In the year 2007, approximately 80% of the whole production has been exported to foreign countries, either by the company itself or by agents.

The first drilling devices were driven on combustion engines and basically constructed for the usage in domestic water supply. Nowadays, branches of several materials, such as plastics, reinforced concrete or different steel type can be created by using different kinds of driving units.

Until the year 2007, over 1700 drilling devices and 140000 TONISCO valves have been supplied.
(www.tonisco.fi)

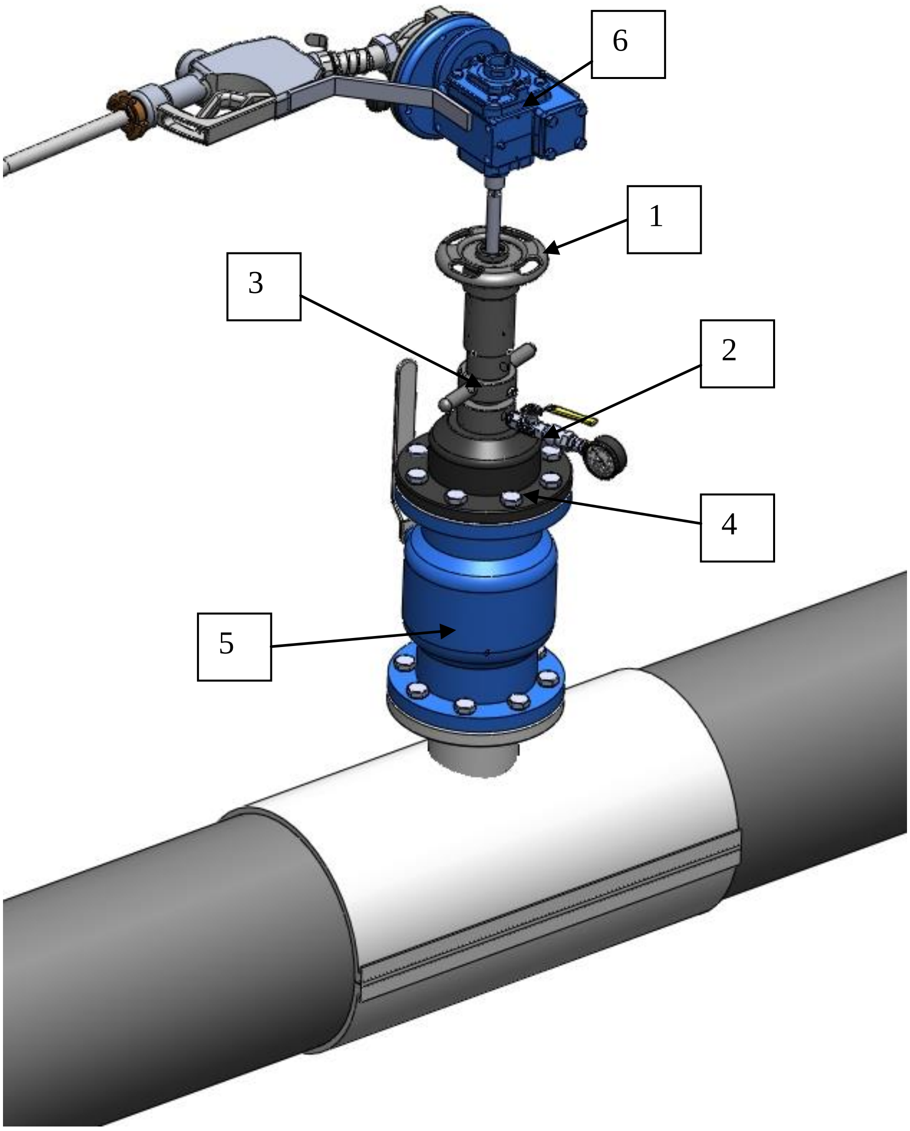


PICTURE 1 Typical TONISCO System Oy product (TONISCO System Oy)

3 HOT TAPPING

3.1 Structure of Hot Tapping Systems

Hot Tapping is the name of a method to create new branches in pipeline systems that are already in use. To create a branch, a valve needs to be attached to the main pipe and an opening has to be cut. Depending on the main pipe material, branches are connected firmly bonded or friction-locked. The drilling device is mounted directly on the attached valve, whereby adapters may have to be used. Certain units, possibly consisting of several parts, need to fulfil particular tasks to ensure a proper working overall system. Hot Tapping can be used in various sectors, as for example urban water supply, gas lines, cooling lines or district heating.



PICTURE 2 Example of a Hot Tapping system (TONSICO System Oy)

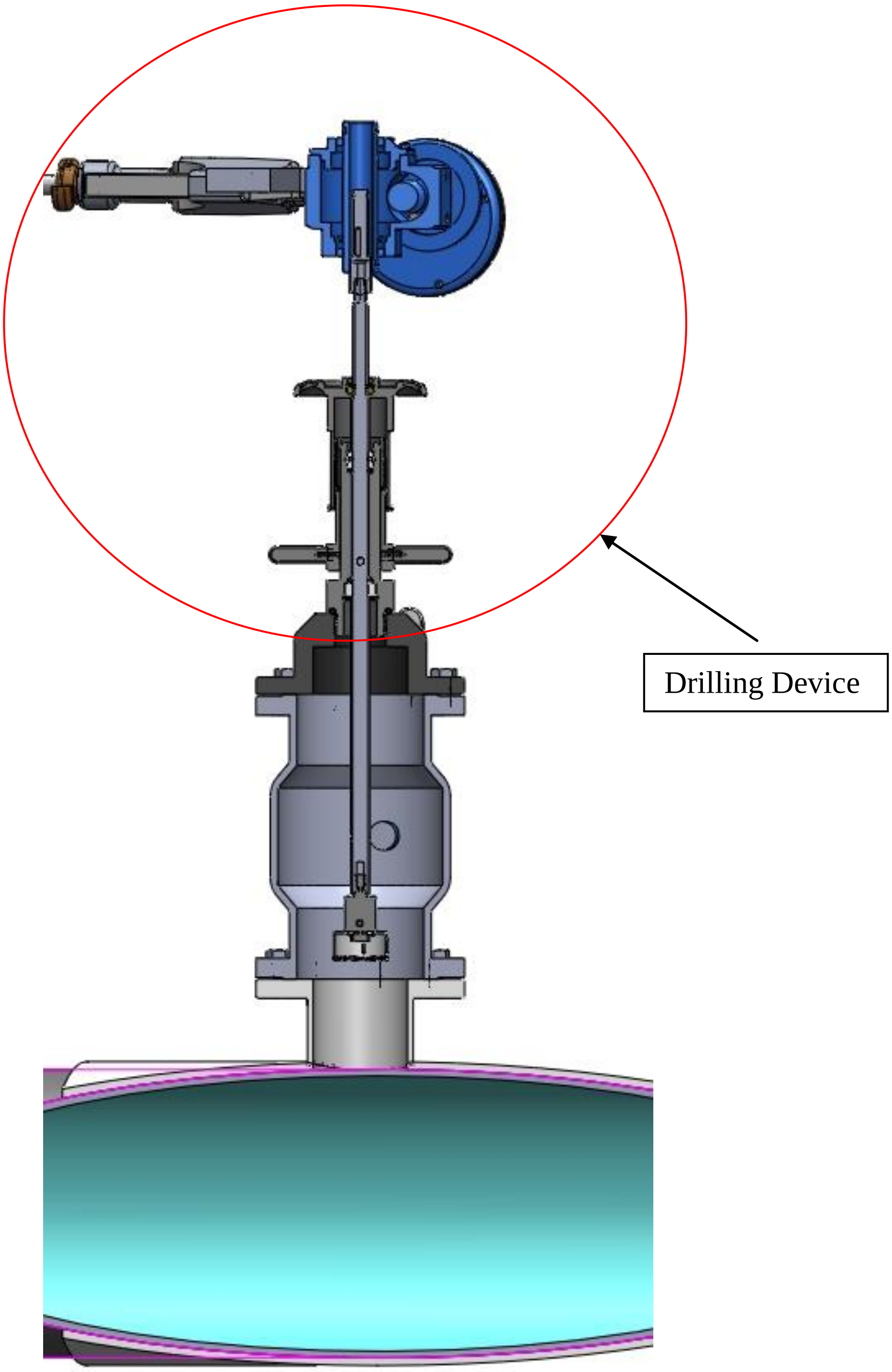
As shown in Table 1 and Picture 2, particular parts can be combined to units which have a specific function for the overall system.

TABLE 1 Units of a Hot Tapping System

	Unit	Task
1	Feed Wheel	Turning the feed wheel clockwise realizes a forward feed of the shaft.
2	Control Tap	Can be used to control the pressure during the drilling by attaching a manometer. It is also possible to use it as a test tap for testing the lock of the valve.
3	Adjusting Socket	Depending on the valve length and on possibly attached extensions, the drilling device can be adjusted in different positions.
4	Adapter	Realizes a suitable connection between the valve and the drilling device. Several types are available to mount the drilling device on different types of valves.
5	Valve	Part of the actual branch that stays in the system and that is attached to the main pipe. For Hot Tapping it needs to have a shut-off device in order to mount or demount the drilling device.
6	Driving Unit	Rotates the shaft and provides a cutting force to the pilot drill and the hole saw.
	Pilot Drill	Used as a guide for the hole saw. There are two hooks mounted on it, so that they can catch the cut circle. The inflow through the pilot hole cools the hole saw with the incoming fluid.
	Hole Saw	Does the actual drilling with the suitable diameter. It is cooled by the fluid inflow thought the pilot hole.
	Shaft Break	Used to ensure a safety release of the adjusting socket.

3.2 Drilling Device

As already mentioned, the drilling device is mounted directly on the valve, whereby a suitable adapter and shaft extensions may have to be used. Since the valves are usually delivered by manufacturers that already have certain certificates in particular fields, only the approval for the drilling device and possibly the adapters is needed.

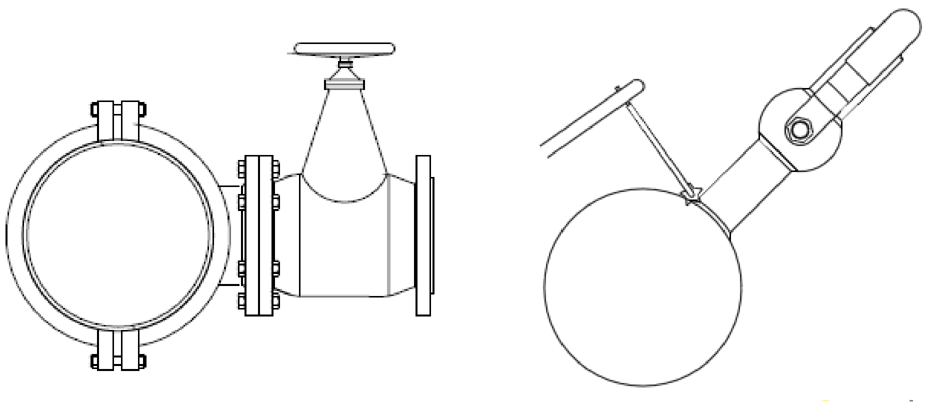


PICTURE 3 Cross-sectional view from the TONISCO Jr. drilling device attached to a ball valve and mounted with an adapter (TONISCO System Oy)

After the device has been mounted as shown in picture 3 and the drilling has been completed, the device is pressurized by the fluid. Thus, the sealings and their tightness have to be proven regarding occurring leakages. For prevention, several O-Rings are placed between the shaft and the body part of the device. A Viton ring out of fluorocarbon elastomeric material can withstand a maximum temperature of 204 °C and so it sets the maximum allowable temperature of the fluid. Other standard parts e.g. retaining rings, axial and guide bearings for the shaft are used in the device as well. (www.o-ring.info) (TONISCO Jr. Manual)

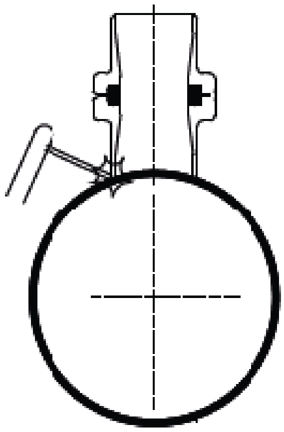
3.3 Valves

A closable valve as a connection between the main pipe and the branch line is necessary for using the drilling device. Since the device has to be assembled and disassembled a shut-off device is mandatory. As shown in picture 4, several valve types are able to be used for branching, whereby it can be necessary to add shaft extensions due to the different valve lengths. Process parameters, such as the possibility to weld, the fluid type and properties and the corrosion of the branch material have influence on the selection of the valve.



PICTURE 4 Flanged valve (l) and ball valve(r) (TONISCO System Oy)

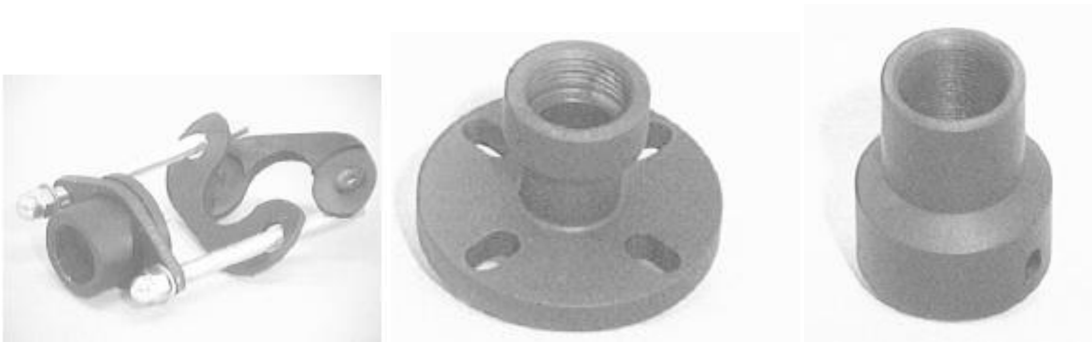
The company TONISCO System Oy supplies also its own valves which are especially designed for the TONISCO drilling devices. TONISCO valves can have different diameters and are welded to the main pipe as shown in picture 5. A sluice plate is acting as a shut-off device.



PICTURE 5 TONISCO Valve (TONISCO System Oy)

3.4 Adapters

Since a connection has to be realized between valve and drilling device, it might be necessary to use adapters. Some examples are shown in picture 6. Depending on the diameter, the valve type and the connection type of the branch, several adapters are able to be attached. Besides the drilling device, it is usually necessary to prove the strength of the adapters as well.



PICTURE 6 Examples for adapters (TONISCO System Oy)

4 EC TYPE-EXAMINATION

4.1 Pressure Equipment Directive

In 1997 the European parliament passed a guideline to harmonize the legal regulations for pressure equipment of the separate European countries. Thus, the Pressure Equipment Directive is valid in the whole European Union.

The PED is an administrative fiat that contains several information about the handling with pressurized equipment e.g. materials, design or manufacturing of pressurized parts. Furthermore, the requirements for the EC Type-Examinations and other approvals (e.g. Product Quality Assurances) are described and a conformity assessment can be done.

4.2 Categorization According to the PED

The Pressure Equipment Directive has to be used for a relative fluid pressure over 0.5 bar. Is this requirement fulfilled, pressure equipment needs to be categorized according to annex II.

It’s possible to define a category by using the conformity assessment tables(annex II), whereby the correct table is chosen by the fluid parameters and the kind of equipment according to article 3.When defining pressurized accessorises which are not directly vessels, pipes or heated pressure equipment, they have to be categorized by the equipment assembled to it (article 3 section 1.4). In the case of Hot Tapping, the accessory (drilling device) is attached to a pipe and is thus covered by the pipe assessment tables.

According to the Pressure Equipment Directive article 9, fluids are divided in two groups:

1. Group 1 describes the dangerous fluids, such as explosive, highly flammable or toxic fluids
2. Group 2 describes all the fluids that are not mentioned in Group 1 (e.g. drinking water)

In addition to the fluid group, the process parameters have to be set to categorize the pressure equipment. These are:

1. Maximum allowable pressure (PS)
2. Maximum/Minimum allowable temperature (TS)
3. Nominal size (DN) or volume (V)

After the definition of all parameter values, the equipment category can be determined by using the suitable conformity assessment table. Four categories are existing, whereby pressure equipment of category I has to fulfil the lowest and of category IV the highest requirements which can be e.g. a Product Quality Assurance (module E) in addition to the EC Type-Examination (module B).

4.3 VdTÜV Regulations

In order to get an approval and a permission to add the CE-Mark on the drilling device, it has to be successfully tested by a notified body. One of these notified bodies that is allowed to conduct an EC Type-Examination is the VdTÜV in Germany.
The EC Type-Examination approval for drilling devices given by the VdTÜV is based on their regulations for valves. As an instruction for the applicant, the VdTÜV created a leaflet called Armatur 100 which is valid in the ambit of the PED.

The Armatur 100 leaflet is applicable for pressure equipment of category III and IV, as well as for category I and II when choosing the module B (EC Type-Examination) for the approval. Pressure equipment categories can be assessed according to Annex II of the PED. Depending on the category, valves may have to fulfil additional requirements to the EC Type-Examination (module B) such as a Product Quality Assurance (module E).

Even though Armatur 100 is basically founded on the determinations of the Pressure Equipment Directive, it contains requirements exceeding it.

4.3.1 **Requirements Regarding the Dimensioning and Design**

According to the Pressure Equipment Directive annex I section 2, the design of pressure equipment can be done by:

- Calculation method and if necessary additional experimental design method
- Experimental design method without a calculation method if PS*DN< 3000 bar or PS*VN<6000 bar*l

The design and dimensioning of valves has to be done according to the regulations of the Pressure Equipment Directive annex I section 2. This section gives among others information about the allowed calculation methods, the resistance, security factors and stability aspects.

Calculation Methods that can be used for the design of pressure equipment are:

- Formulas (e.g. according to equations from standards)
- Analysis (e.g. FEM analysis)
- Fracture mechanics

Generally, "…the allowable stresses for pressure equipment must be limited having regard to reasonably foreseeable failure modes under operating conditions…".(Pressure Equipment Directive)

Since several standards (e.g. DIN EN standards) are fulfilling the regulations of the PED concerning all aspects, it is highly recommended to use them. Nevertheless, if no exact standard is available, the dimensioning can be made according to other sources while it is compulsory to take every point of the PED into account. Armatur 100 demands an additional consideration of the DIN 12569, if necessary.

DIN EN 12569 is a European standard that gives information about valves DN15 > in chemical or petrochemical process industry. Besides that, the VdTÜV listed certain other standards (DIN EN...) and policies (AD 2000) in the Armatur 100 leaflet which can be seen as basic test principles. Even though it is highly recommended to use them, it is not stringently required and other standards are allowed to be used, if they fulfil the requirements of the Pressure Equipment Directive annex 2.

4.3.2 Provision of Documents for the EC Type-Examination

A contact person who gives instructions and feedback needs to be found before the whole application process can start. Next to some general details such as the name and address of the applicant, certain technical documents have to be submitted to the contact person of the VdTÜV. These are listed in the leaflet Armatur 100.

TABLE 2 Documents for the VdTÜV approval

	Required Documents according to Armatur 100
1	General description of the EC Type-Examination part to be approved
2	Details of the valve such as: 1. Categorization of the valve according to the PED 2. Type reference 3. Size (nominal size) 4. Operating pressure 5. Operating and ambient temperature 6. Used fluid 7. Purpose of use 8. Specialities
3	Assembly drawing and if necessary part drawings
4	Explanations for the comprehension of the drawings of the valve
5	Material details
6	Report of the design and dimensioning of the valve (Strength calculations)
7	List of the used standard listed in the PED article 5 or explanations of another suitable standard
8	Details to conducted tests during or after the production
9	Details to the required qualifications according to the Pressure Equipment Directive annex I
10	Risk analysis according to the Pressure Equipment Directive annex I
11	Manual according to the Pressure Equipment Directive annex I

The whole certification is a quite protracted process and thus only the provisions of point 2, 6 and 11 in the upper list are part of this thesis.

4.3.3 **Documents to Provide as Part of the Thesis Work**

The creation of a project management plan as an approach for the obtaining of certificates for the TONISCO Jr. and other TONISCO products needed to be done. In the plan, the required steps are visualized, whereby they can vary depending on the fluid category. In appendix I, the project management plan is added.

As a part of the thesis work, a user manual needed to be translated from English to German, since the VdTÜV is a German organisation. Appendix II shows a part of this translated manual, whereby the original manual is much more extensive. In general, user manuals for pressure equipment needs to be done according to the PED annex I section 3.4.

As the original documents of the strength calculations are quite old, the provision of new ones is a very detailed part of this thesis, whereby TONISCO System Oy has to decide if they use them. Strength calculations can only be seen as an examination and documentation of the pressurized parts, since the responsibility of the correctness haveto be carried by the authorized person(s).

4.4 Steps after the Provision of Documents

After all required documents have been provided, the applicant company and the notified body are fixing a date for an examiner to accomplish the test. Besides the inspection of the documents and the comparison of the real model with the drawings, the applicant company has to provide additional verifications before the test can start.
These are:

- Calibration of the measurement systems
- Materials according to DIN 10204- 3.1

The test procedure is shown in the leaflet Armature 100 and will not be explained in this thesis due to the fact that it is an extensive issue.

After all tests have been successful and executed according to certain DIN standards, the test certificate is handed out and the product can be marked with a CE-mark.

4.5 Categorization of the TONISCO Jr. Drilling Device

The VdTÜV requires a categorization of the drilling device according to the Pressure Equipment Directive, which can be done as shown in chapter 4.2 of this thesis.

The first approval to acquire should be the one for district heating. Fluid in district heating systems can be defined as fluids of group 2 (not dangerous) according to the Pressure Equipment Directive article 9. Furthermore, the process parameters for the TONISCO Jr. are set by the applicant company TONISCO System Oy.

1. Maximum allowable pressure (PS) = 40bar
2. Maximum allowable temperature (TS) = 200°C
3. Nominal size (DN) = 100mm

In order to use the correct conformity assessment table, the suitable section and intend has to be found in article 3 of the PED. Certain values have to be calculated or taken from tables to be compared with the limit values. Since the drilling device is attached to a pipe, it has to be handled as one of those (PED article 3, section 1.3).

$$PS * DN = 40\ bar * 100 = 4000\ bar$$

Vapour pressure of water at t = 200 °C

$$p_{v200°C} = 15.544\ bar$$

(http://anorganik.chemie.vias.org/dampfdruck_tabelle_wasser.html)

Since $PS * DN = 4000\ bar > 1000\ bar$ and $p_{v200°C} = 15.544\ bar > 0.5\ bar$, section 1.3(b), intend 2 is an accurate description for the classification of TONISCO Jr.. As referred in the paragraph, diagram 7 of annex II of the Pressure Equipment Directive can be used. From the conformity assessment table it is possible to determine the TONISCO Jr.as pressure equipment of category I. For that reason, an EC Type-Examination is enough for the approval and the obtaining of the CE-Mark.

5 STRENGHT CALCULATION OF THE TONISCO JR DRILLING DEVICE

5.1 General Notes

Strength calculations have to be done very precisely since the manufacturing company is responsible in case of accidents or damages. For that reason, the calculations in this thesis need to be seen more as an examination, whereby the company TONISCO System Oy has to decide if they are using the results and calculation ways.

To be consistent with the Pressure Equipment Directive and the additional requirements from the VdTÜV, it is recommended to use standards for the dimensioning, as most of them are fulfilling the PED requirements. Certain standards are listed in the Armatur 100 leaflet, while only part of them can be used for the drilling device, due to the fact, that the device is not a real valve as the leaflet is actually made for. Furthermore, it has to be considered, that the product is already in use for a long period of time without any problems and standards are quite often made for the design of products.

In many standards (e.g. DIN EN 12516-2), calculation methods for the minimum thickness of walls or the height of lids is shown for different forms and shapes of valves and other pressure equipment. Since no listed shape can be used to calculate the drilling device and the product is already manufactured many years, the dimensioning of the walls and lids is not necessary anymore. It is more reasonable to analyse the maximum stresses in the device while using characteristic material values from standards and reliable tables.

As the main focus lays on the approval for district heating, the DIN EN 12569 for chemical and petrochemical industry is not decisive for this approval.

5.2 Resistance of Pressure Equipment

The PED requires the consideration of certain aspects for the dimensioning of pressure equipment, listed in annex I section 2. Table 3 explains the kind of demand and how it concerns the drilling device.

TABLE 3 Demands to be taken into account (according to the PED)

Demand	Relation to the Drilling Device and Consideration of the demand
Calculation pressure is not allowed to be lower than the maximum allowable pressure	PS = 40bar = maximum allowable pressure
Consideration of all temperature and pressure combinations	Maximum allowable pressure PS = 40 bar in combination with the maximum temperature TS = 200 °C
Maximum stress has to be lower than the maximum allowable stress	$\sigma_{existing} \leq \sigma_{allowed}$
Appropriate safety factors	Safety factor SF = 1.5 (given by the VdTÜV)
Yield strength 0.2 or 1.0 of proof strength at calculation temperature	Taken from reliable tables (e.g. DIN EN 10269)
Tensile Strength	Taken from reliable tables
Time depend strength e.g. creep strength	Since the drilling device is just in use for a small period of time, not decisive
Fatigue data	Since the drilling device is just in use for a small period of time, not decisive
Modulus of elasticity	Depends on the material. Since mostly steel is used in the drilling device, E ≈210000N/mm²
Appropriate amount of plastic strain	If below the maximum allowed stress, no plastic strain should occur. In notches, a possible plastic deformation may needs to be taken into account by formulas.
Impact strength and fracture toughness	Not decisive due to the high temperature. Usually construction steel and steel for tempering is used which is quite ductile
Creep, fatigue and corrosion	Since the drilling device is just in use for a small period of time, not decisive

5.3 Theoretical Background (Thick walled pipes)

Since the drilling device can be seen as a kind of pressure vessel or pipe, the occurring pipe wall stress needs to be calculated.

Is the relation between the inner and the outer radius of a pipe relativly high, it is not possible to anticipate a uniform stress distribution in the pipe wall anymore (as in thin pipe walls). Furthermore, the size of the middle radius has to be taken into account whereby the wall thickness usually increases with a rising size of the radius. To calculate e.g. thick walled vessels, hubs or discs it is necessary to use another mathematical approach.

Cutting a very small disc out of a cylinder wall,it is possible to define the forces that are acting on an infintesimal small element of the wall. After that, the balance of forces according to d'Alembert can be formed on the system whereby cylinder coordinates should be preferably used.

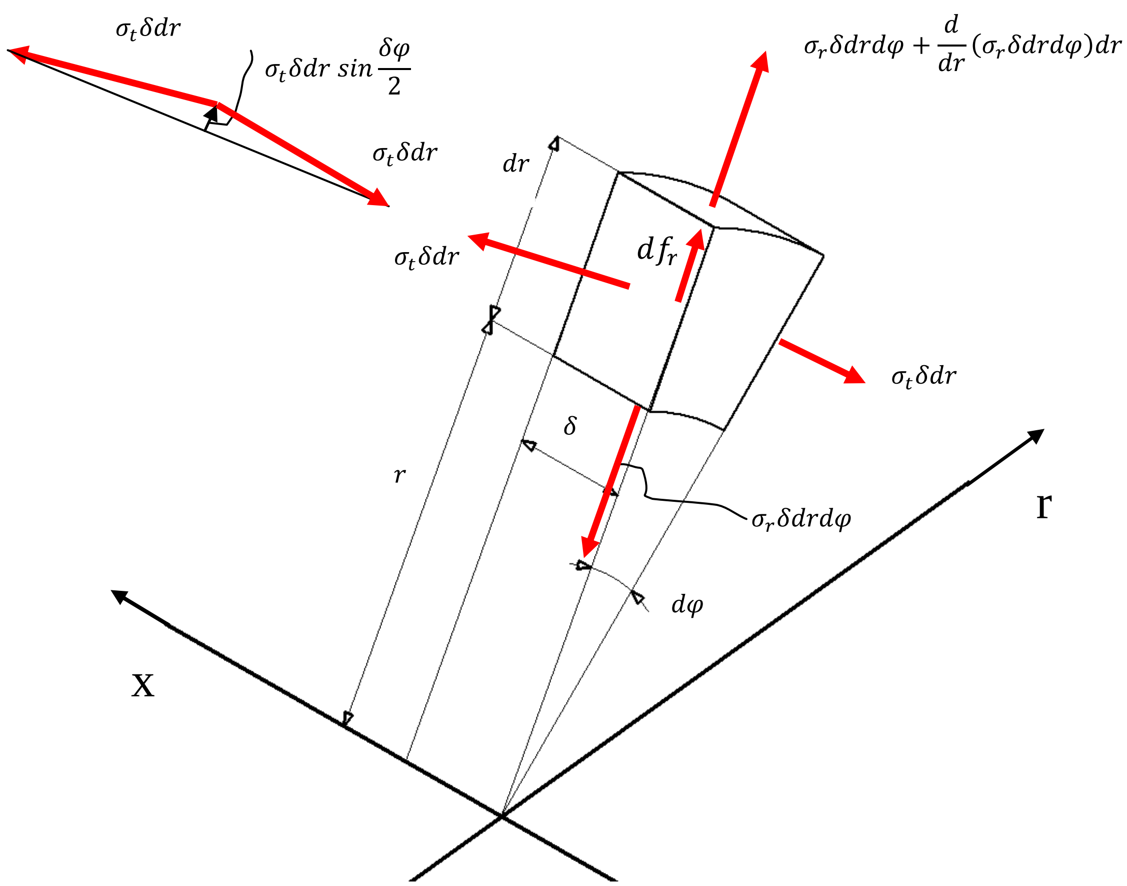


FIGURE 1 Forces inside an infinitesimal small element of a pipe wall

Figure 1 shows the occurring forces. The balance of forces can be written in the direction of the radius.

$$\sum F_r = -\sigma_r \delta dr\, d\varphi + \sigma_r \delta dr d\varphi + \frac{d}{dr}(\sigma_r \delta dr d\varphi) dr + \sigma_t \delta dr sin \frac{d\varphi}{2} + df_r = 0 \qquad (1)$$

For pipe walls the volume forces can be determined as $f_v = 0$.
With $sin \frac{d\varphi}{2} \approx \frac{d\varphi}{2}$ the equation can be simplified.

$$\frac{d}{dr}(\sigma_r \delta dr d\varphi) dr - \sigma_t \delta dr \frac{d\varphi}{2} + df_r = 0 \qquad (2)$$

After using the product rule and doing some transformations, the stress equation of the thick walled pipe can be visualized as:

$$r \frac{d\sigma_r}{dr} + \sigma_r - \sigma_t = 0 \qquad (3)$$

Due to the fact that this differential equation has two different, unknown stress types it is impossible to solve it. Therefore, another equation that shows dependencies between both stress forms (σ_r and σ_t) is needed. The dependencies between these two stresses can be described by using the strains (ε_r and ε_t), the law of Hook and the transversal contraction (ν).

Equivalent to the balance of forces, the compatibly condition is built, with the solution

$$r \frac{d\varepsilon_r}{dr} + \varepsilon_r - \varepsilon_t = 0 \qquad (4)$$

With the help of the descriptions of strains,

$$\varepsilon_r = \frac{1}{E}(\sigma_r - \nu \sigma_t) \text{ and } \varepsilon_t = \frac{1}{E}(\sigma_t - \nu \sigma_r) \qquad (5)$$

the second equation can be formed. In there, E can be eliminated and the equation can be simplified as:

$$r\left(\nu\frac{d\sigma_r}{dr} - \frac{d\sigma_t}{dr}\right) + (1+\nu)(\sigma_r - \sigma_t) = 0 \tag{6}$$

After having two equations with two unknown factors, it is possible to solve the equation by using the substitution method and defining the external influence conditions.

1. $r_i = r$ =>$\sigma_r = -p_i$

   and

2. $r_a = r$ =>$\sigma_r = 0$

The differential equation can be solved by using the mathematical approach $\sigma_r = Cr^n$. After defining the constants C_1 and C_2 and doing some other transformations, the solution can be visualized as:

$$\sigma_r = -p_i * \frac{{r_i}^2}{({r_a}^2 - {r_i}^2)}(\frac{{r_a}^2}{r^2} - 1) \tag{7}$$

$$\sigma_t = p_i * \frac{{r_i}^2}{({r_a}^2 - {r_i}^2)}(\frac{{r_a}^2}{r^2} + 1) \tag{8}$$

To transform the multi-axial stress condition into a uniaxial stress condition, the shape modification hypothesis (according to von Mises) or the shear stress hypothesis can be used.

$$\sigma_v = \sqrt{\frac{1}{2}[(\sigma_r - \sigma_t)^2 + (\sigma_t - \sigma_l)^2 + (\sigma_l - \sigma_r)^2]} \tag{9}$$

or

$$\sigma_v = \sigma_{max} - \sigma_{min}$$

(DIN 2413.1)

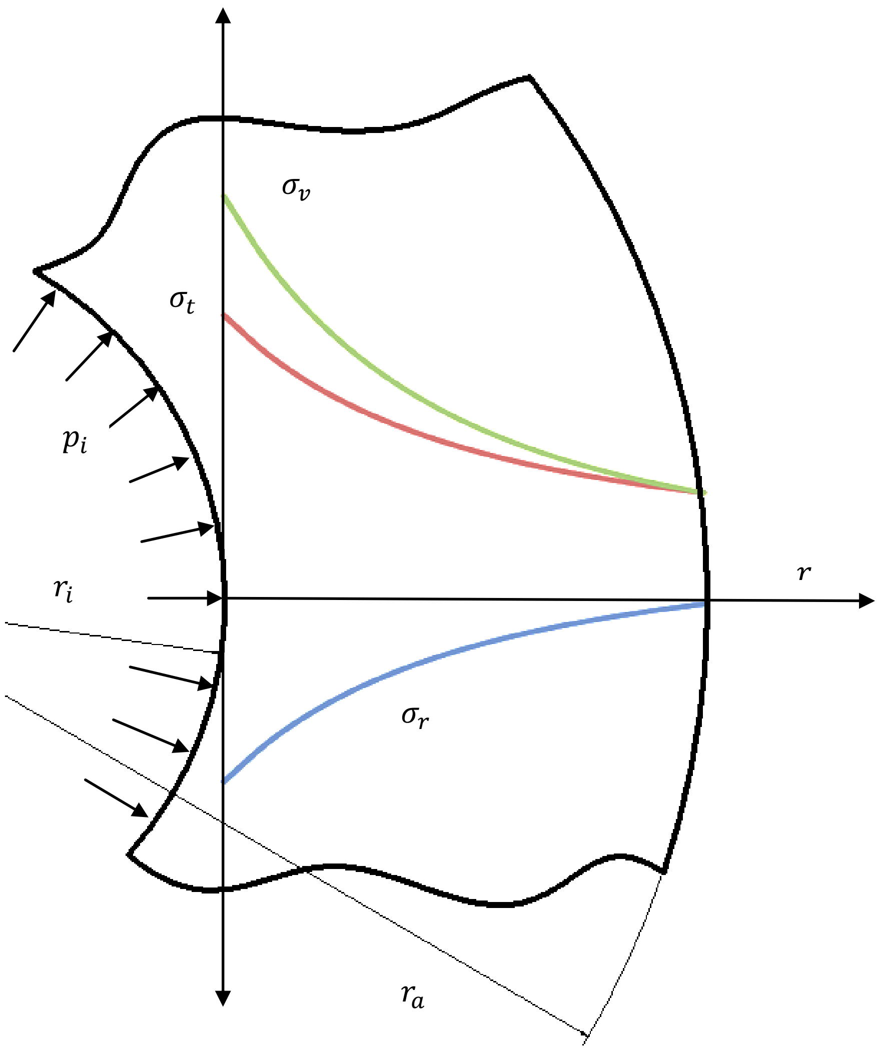


FIGURE 2 Stress distributions in a thick pipe wall (shape modification hypothesis)

Figure 2 shows the different stress curves in a thick walled pipe which is pressurized from the inside without longitudinal stresses. While the equivalent stress is the highest at the inside of the pipe wall, it gets lower when the diameter increases. (Technische Mechanik Festigkeitslehre)

[Equation (1) to (9) were taken from Technische Mechanik Festigkeitslehre]

5.4 Load Assumption and Materials

The load assumption needs to include the stress of the drilling process and of the inner fluid pressure, whereby the temperature has to be taken into account as well. Since the test pressure is 1.5 times higher than the maximum pressure a security factor of SF= 1.5 must be used for all materials.

TONISCO Jr. drilling devices have to resist a maximum pressure caused by the fluid of $p = 40$ bar at a temperature of $t_i = 200°C$.

The following parameters have been used for the calculations:

$p_i = 40\text{bar} = 4\ N/mm^2$
$t_i = 200°C$
Fluid medium: water

As the most endangered parts and connections, the following points have been proven.

- Thread in Part 10
- Weakest point in Part 10
- Force screws, Part 9
- Thread in Part 6 and 1
- Weakest point in Part 6
- Retaining ring in Part 1
- Bearing in Part 1
- Weakest point in Part 1

The maximum allowable stress of a component depends on its material. A security factor of SF = 1.5 has to be divided from every yield strength value.

$f = R_{p0,2/t}/SF$ (DIN EN 12516-2, page 8)

TABLE 4 Material Strengths

Part	Material	Yield strength at temperature of 20°C [N/mm”]	Yield strength at temperature of 200°C [N/mm^2]	Maximum allowable stress [N/mm²]
10	42CrMo4	-	640	426
6	S355 J2G3	355	-	236
1	S355 J2G3	355	-	236
9	42CrMo4	900	-	600

Material data can be taken from the DIN EN 10269 (page 21) as well as from the Roloff/Matek Tabellenbuch (page 1)

5.5 Part 10, Body

Part 10 is one of the few parts of the device that have a direct contact with the fluid. The pressure formed by the water or gas creates a force in the direction of the pipe wall and the shaft (x- direction). The thread has to carry the entire load coming from the inner fluid pressure in x- direction plus an additional load from the drilling force F_f. Because all sites of the pipe wall are stressed equally, the forces will annul each other and no moments will result. The small part of the control tap can be neglected. Since the fluid can also have a temperature up to 200 °C, a weakening of the material has to be taken into account.

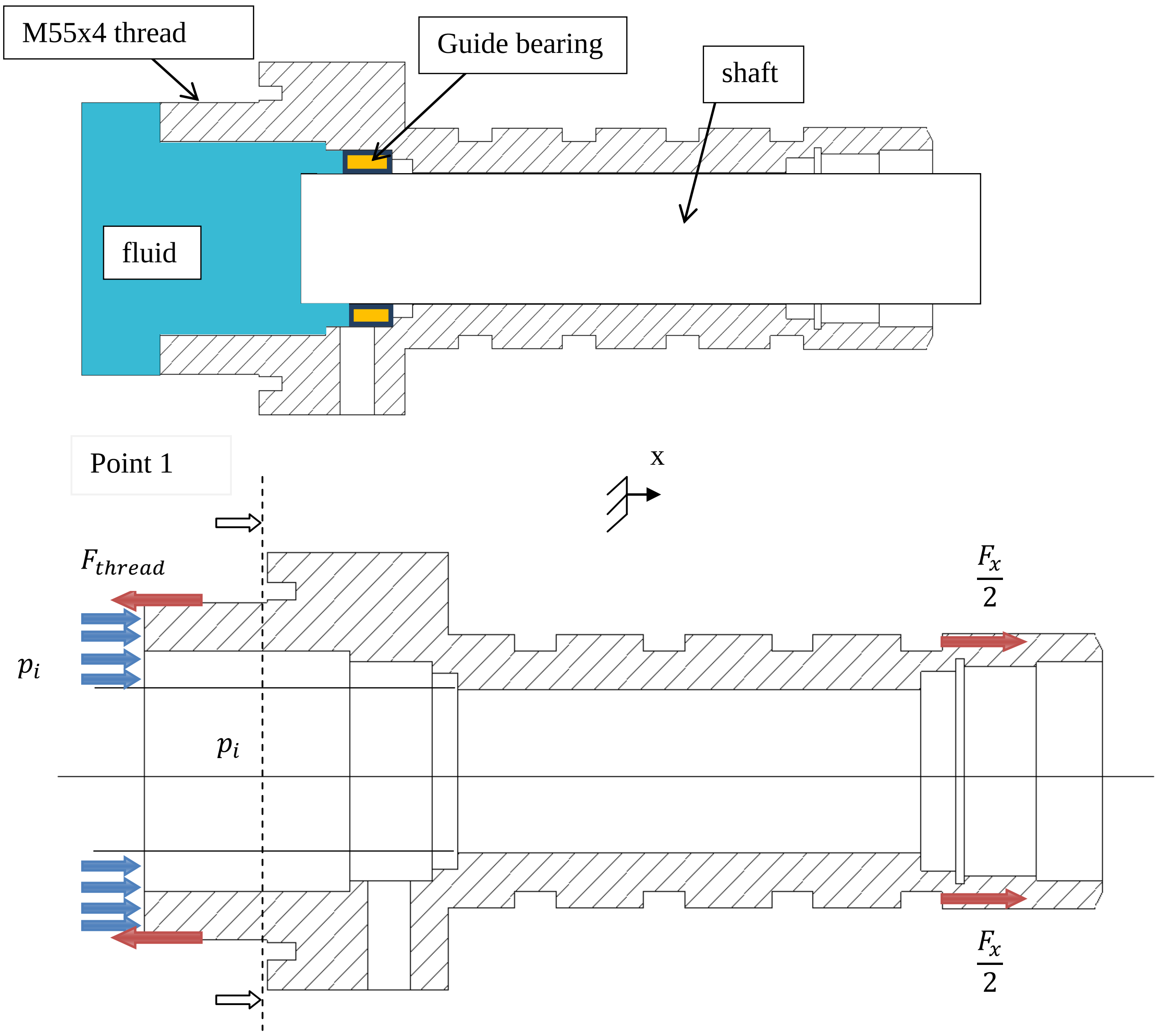


FIGURE 3 Structure and forces in Part 10 (x-direction)

In figure 3, the occurring forces in Part 10 are visualized. The force F_x is initiated into the shaft and is later returned to the part as a sum of the fluid pressure and the feed force caused by the drilling process.

$$F_x = p_i * A_s + F_f$$
$$F_x = F_p + F_f$$

The area of the shaft A_s is calculated by:

$$A_s = \frac{\pi * {d_s}^2}{4} = \frac{\pi * (21.5\ mm)^2}{4} = 363.05\ mm^2$$

From this it follows that the shaft force caused by pressure can be calculated as:

$$F_p = A_s * p_i = 363.05\ mm^2 * 4\ N/mm^2 = 1452.2\ N$$

The drilling force F_f has to be estimated since the forces are varying with the feed rate and the condition of the driller. The specific cutting values are experimentally determined and they highly depend on the main pipe material and the feed rate.

$$F_f = 2000\ N$$

The overall force in x-direction can be determined as the sum of both forces.

$$F_x = 1452.2\ N + 2000\ N = 3452.2\ N$$

As visible in Figure 3, the guide bearing inside Part 10 inducts an additional force in x-direction coming from the fluid. An annulus with the diameters $d_{ibearing} = 21.5\ mm$ and $d_{obearing} = 35\ mm$ can be defined as the cross section.

$$A_{bearing} = \frac{\pi * ({d_{obearing}}^2 - {d_{ibearing}}^2)}{4} = \frac{\pi * [(35\ mm)^2 - (21.5\ mm)^2]}{4}$$

$$A_{bearing} = 599.06\ mm^2$$

$$F_{bearing} = A_{bearing} * p_i = 599.06\,mm^2 * 4\,N/mm^2 = 2396.25\,N$$

Since the drilling device is screwed in either the adapter or directly to the TONISCO Valve, the pressurized fluid acts on the front cross-section as well. The dimensions of the annulus cross section can be determined with $d_{ivalve} = 35\,mm$ and $d_{ovalve} = d_{thread} = 55\,mm$.

$$A_{valve} = \frac{\pi * ({d_{ovalve}}^2 - {d_{ivalve}}^2)}{4} = \frac{\pi * [(55\,mm)^2 - (35\,mm^2)]}{4} = 1413.72\,mm^2$$

$$F_{valve} = A_{valve} * p_i = 1413.72\,mm^2 * 4\,N/mm^2 = 5654.88\,N$$

The M55x4 thread has to carry all the forces in x-direction, including the loads that affect the shaft (fluid and feed force in x-direction). It can be calculated as a sum of all part forces or as a circle formed area with the outer diameter of the thread multiplied by the fluid pressure plus the feed force.

$$F_{thread} = \frac{\pi * {d_{thread}}^2}{4} * p_i + F_f = \sum_i F_i$$

$$F_{thread} = \frac{\pi * (55\,mm)^2}{4} * 4\,N/mm^2 + 2000\,N = 11503.3\,N$$

5.5.1 **Weakest Point**

Besides the shaft, the guide bearing and the retaining ring, Part 10 is the only metallic part that has a contact with the fluid and hence, it is exposed to high temperatures. Assumed that Part 10 is heated up to the maximum temperature of $t_i = 200°C$, which is not a realistic scenario, the yield strength of the material has to be considered for that temperature.

In order to find the weakest point in Part 10, factors like the location of the maximum force or the different cross sections have to be observed. If the contact areas of the screws are disregarded, the point can be found at the end of the thread at the location of the undercut. The weakest point at the location of the undercut (point 1), has diameter dimensions of $d_{i1} = 37mm$ and $d_{o1} = 51\ mm$. According to DIN 2413 and the AD-leaflets a vessel is defined as a thin vessel when the diameter ratio is ${d_o}/{d_i} \leq 1.2$.

Because of ${d_{o1}}/{d_{i_1}} = {51mm}/{37mm} = 1.378 \geq 1.2$ the formulas for thick vessels should be used. The highest stress in the direction of pipe wall is located at the inside wall of the pipe, which means that the inner radius can be inserted in equation 7 and 8 of section 5.3.

$$\sigma_r = -4\ {N}/{mm^2} * \frac{(18.5\ mm)^2}{((25.5\ mm)^2 - (18.5\ mm)^2)}\left(\frac{(25.5\ mm)^2}{(18.5\ mm)^2} - 1\right) = -4\ N/mm^2$$

$$\sigma_t = 4\ {N}/{mm^2} * \frac{(18.5\ mm)^2}{((25.5\ mm)^2 - (18.5\ mm)^2)}\left(\frac{(25.5\ mm)^2}{(18.5\ mm)^2} + 1\right) = 12.89\ N/mm^2$$

To calculate the stress σ_{x1} at the location of point 1 in figure 3, the force in x-direction has to be divided by the load-bearing cross section. An equal stress distribution over the whole section is assumed.

$$A_{x1} = \frac{\pi * ({d_o}^2 - {d_i}^2)}{4} = \frac{\pi * (51\ mm^2 - 37\ mm^2)}{4} = 967.61\ mm^2$$

$$\sigma_{x1} = \frac{F_x + F_{bearing}}{A_{x1}} = \frac{(2396.25\ N + 3452.2\ N)}{967.61\ mm^2} = 6.04\ {N}/{mm^2}$$

According to DIN 2413, the shape modification hypothesis according to von Mises (section 5.3equation 9) can be used to estimate the state of stress.

$$\sigma_v = \sqrt{\frac{1}{2}[(-4N/mm^2 - 11.08N/mm^2)^2 + (11.08N/mm^2 - 4.813\ N/mm^2)^2 + (4.813\ N/mm^2 + 4N/mm^2)^2]}$$

$$\sigma_v = 14.71\ {N}/{mm^2}$$

As another approach, the maximum shear hypothesis is allowed to be used.

$$\sigma_v = \sigma_{max} - \sigma_{min} = 12.89\ N/mm^2 - (-4\ N/mm^2) = 16.89\ N/mm^2$$

As listed in Table 4 section 5.4, the material of Part 10 (42CrMo4) has a maximum allowable stress of

$$f_{allowable} = 426\ N/mm^2$$

$\Rightarrow$ $\quad 13.12\ N/mm^2 \ll 426\ N/mm^2$

or $\quad 16.89\ N/mm^2 \ll 426 N/mm^2$

5.5.2 **Flank Pressure of the thread M55x4**

Due to the fact that the connection just exists for a small period of time and will be removed after the drilling, the maximum flank pressure of the thread is decisive for the design. The maximum pressure the thread can withstand is estimated as $p_{max} = 12.5\ N/mm^2$(Roloff/Matek Tabellenbuch, page 107). Divided by the security factor, the maximum allowable flank pressure is determined as:

$$p_{max} = \frac{12.5\ N/mm^2}{SF} = \frac{12.5\ N/mm^2}{1.5} = 8.3\ N/mm^2$$

$$p_{flank} = \frac{F_{thread} * P}{l_1 * \pi * d_2 * H_1}$$

(Roloff/Matek Formelsammlung, page 81)

$l_1 \approx 18mm\ ; H_1 = 0.54127 * P = 0{,}54127 * 4 = 2.165mm$

(Roloff/Matek Tabellenbuch, page 91)

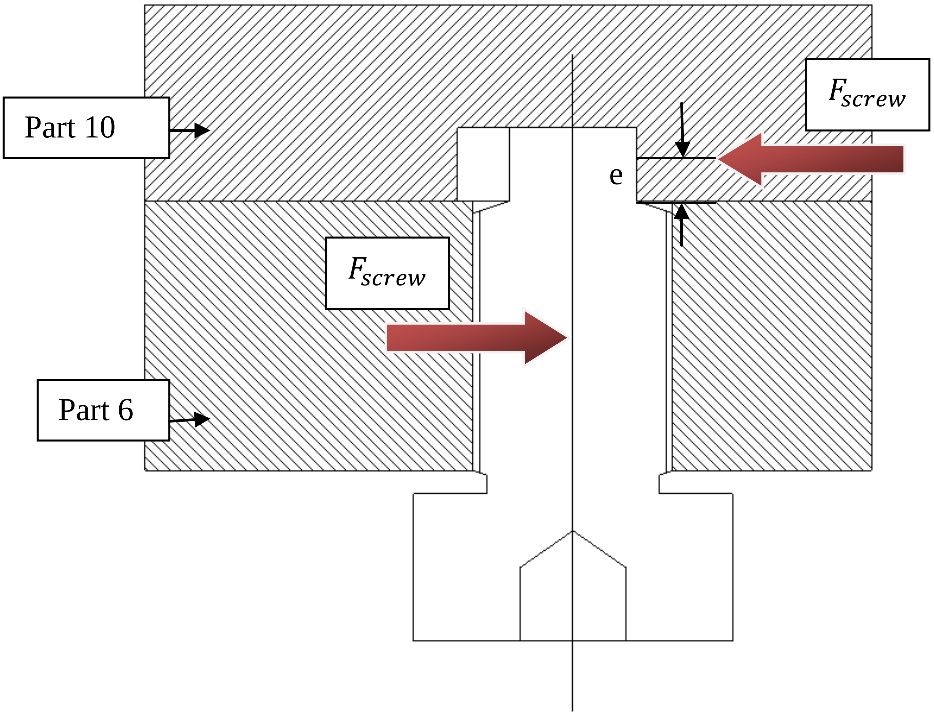
$d_2 = 52.402mm$

$$p_{flank} = \frac{11503.31\ N * 4\ mm}{18\ mm * \pi * 52.402\ mm * 2.165\ mm} = 7.17 \frac{N}{mm^2} \leq 8.3 N/mm^2$$

5.6 Part 9, Power Screws

In steel constructions, screw connections have to be proven on shearing, hole bearing and the weakening of the cross section. In this case, the shearing is not enough to determine the mechanical stress in the screws. The decisive parameter is a high moment that occurs, due to the weak cross section and the relatively high force. Both screws (n=2) of the device are carrying half of the load in x-direction and are stressed equally. Hole bearing is also calculated, whereby it is proved in Part 6, since it is not possible to ascertain it in Part 10 with the conventional formulas. Moreover, the material of Part 6 is weaker.

$$F_{screw} = \frac{F_x}{n} = \frac{F_x}{2} = \frac{3452.2N}{2} = 1726.1\ N$$



FIRGURE 4 Forces affecting the screws

5.6.1 Prove of Stress

The screws are made of 42CrMo4 with a maximum allowable stress of $f_{allowable} = 600\ N/mm^2$ at 20 °C, listed in table 4 section 5.4 of this thesis.

The screws have the minimum cross section at the diameter of $d_s = 5.5mm$

$$A_s = \frac{\pi * {d_s}^2}{4} = \frac{\pi * 5.5\,mm^2}{4} = 23.76\,mm^2$$

Since the distribution of the shear stress is parabolic, the highest shearing stress caused by the force F_{screw} is located in the middle of the circle cross section (no normal forces at this location).

$$\tau_{shear} = \frac{4\,F_{screw}}{3\,A_s} = \frac{4 * 1726.1\,N}{3 * 23.76\,mm^2} = 96.86\,N/mm^2$$

(Technische Mechanik Festigkeitslehre, page 245)

$$\tau_{allowable} = \frac{f_{allowable}}{\sqrt{3}} = \frac{600\,N/mm^2}{\sqrt{3}} = 346.41\,N/mm^2$$

(shape modification hypothesis)

$$96.86\,N/mm^2 \leq 346.41\,N/mm^2$$

The additional moment occurs due to the lever arm e. As a mathematical approach, the system can be seen as a cantilever beam with the highest moment at the end of the small diameter. Nevertheless, the validity of this approach has to be reviewed, as the ratio of beam length to the diameter is quite low. For that, FEM can possibility be used (as shown in chapter 6).

In beams with circle cross sections, the highest normal stress is located at the perimeter.

$$W_b = \frac{\pi * d^3}{32}$$

(Roloff/Matek Tabellenbuch, page 122)

$$W_b = \frac{\pi * (5.5\,mm)^3}{32} = 16.33\,mm^3$$

$$M = F_{screw} * e = 1726.1N * \frac{3.5mm}{2} = 3020.675\,Nmm$$

$$\sigma_b = \frac{M}{W_b} = \frac{3020.675\ Nmm}{16.33\ mm^3} = 184.98\ N/mm^2$$

Because of the notch effect between the diameters of the screw, the highest stress can hardly be determined exactly. As an approach, experimental and analytical methods have been used to visualize the influence of the notch geometry in graphs. From this graphs it can be seen that the size of the notch radius is responsible for the height of the maximum stress peaks. As it depends highly on the used forming tool, the radius has to be estimated.

$$r_{notch} = 0.2\ mm$$

The notch factor can be taken from a graph, after calculating certain values.

$$\frac{r}{d_s} = \frac{0.2\ mm}{5.5\ mm} = 0.036$$

For the bigger diameter, the pitch diameter of the M8 thread is used.

$$D = 7.188\ mm$$

$$\frac{D}{d_s} = \frac{7.188\ mm}{5.5\ mm} = 1.307$$

$$=\gg \alpha_k \approx 2.4$$

(Roloff/Matek Tabellenbuch, page51)

The highest stress in the notch can be determined by:

$$\sigma_{max} = \alpha_k * \sigma_b = 2.4 * 184.98\ N/mm^2 = 443.95\ N/mm^2$$

(Roloff/Matek Maschinenelemente, page 53)

Sometimes (not in this case), local stress peaks caused by the notch are exceeding the maximum allowable stress of the material. These local plastic deformations are not causing a destruction of a component and they can be taken into account by using a plastic support factor.

$$n_{pl} = \sqrt{\frac{R_{pmax}}{R_p}} \qquad \text{with } R_{pmax} = 1050\ N/mm^2$$

(Roloff/Matek Formelsammlung, page 11)

Furthermore, the plastic deformation factor without a notch should be taken from a table.

$$\alpha_{bp} = 1.7$$

(Roloff/Matek Tabellenbuch, page 47)

$$n_{pl} = \sqrt{\frac{1050\ N/mm^2}{900\ N/mm^2}} = 1.08 \leq \alpha_{bp}$$

$$1.08 \leq 1.7$$

As plastic support factor is lower than the plastic deformation factor without a notch, it has to be used, since it is the smaller value.

$$\sigma_{plallowed} = f_\sigma * R_p * n_{pl} \qquad \text{with} f_\sigma = 1$$

$$\sigma_{plallowed} = 600\ N/mm^2 * 1.08 = 648\ N/mm^2$$

$$443.945\ N/mm^2 \leq 648\ N/mm^2$$

(Roloff/Matek Formelsammlung, page 11-12)

The maximum stress of the part is lower than the maximum allowable stress with plastic deformation, which means, that the part is safe when using the assumptions that have been made. It can be said, that these calculations are containing inaccuracies in that way, that unrealistic load assumptions regarding the feed force of the drilling device have been made. This causes different stress values in x-direction. Furthermore, the force varies a lot by changing the radius of the notch and an idealistic geometry has been assumed. Since the real stress distribution can hardly be determined and the peak value is quite high, the part is recommended to be examined with FEM as it is done in section 6.4.

5.6.2 **Prove of Hole Bearing**

The hole bearing is proved in Part 6 because it is has a through hole and is thus possible to ascertain with the conventional formulas. Moreover, the material of Part 6 is weaker than the material of Part 9.
The maximum hole bearing pressure is determined by:

$\sigma_{hbmax} = \alpha_1 * {f_{allowed}}/{S_m}$ whereby $\alpha_1 = 0.73 * \frac{e_1}{d} - 0.2$

with $e_1 = 10\,mm$ $S_m = 1.1$
and $d = 8\,mm$

$$\alpha_1 = 0.73 * \frac{10mm}{8mm} - 0.2 = 0.7125$$

$$\sigma_{hbmax} = 0.7125 * \frac{236\,N/mm^2}{1.1} = 152.86\,N/mm^2$$

The hole bearing pressure caused by the screw force is calculated as:

$\sigma_{hb} = \frac{F_{screw}}{t*d}$ with $t = 14\,mm$ and $d = 8\,mm$

$$\sigma_{hb} = \frac{1726.1N}{14\,mm * 8\,mm} = 15{,}41\,N/mm^2$$

$$15{,}41\,N/mm^2 \leq 152.86\,N/mm^2$$

(Roloff/Matek Formelsammlung, page 75-76)

5.7 Part 6, AdjustingSocket

Since Part 6 doesn’t have a direct contact to the fluid and is thus not pressurized directly by it, only the forces in x direction coming from the screws are acting on Part 6 as it can be seen in figure 5. Furthermore, the fluid temperature doesn’t play a big role, since there is no direct fluid contact and the device is just used for a short period of time.

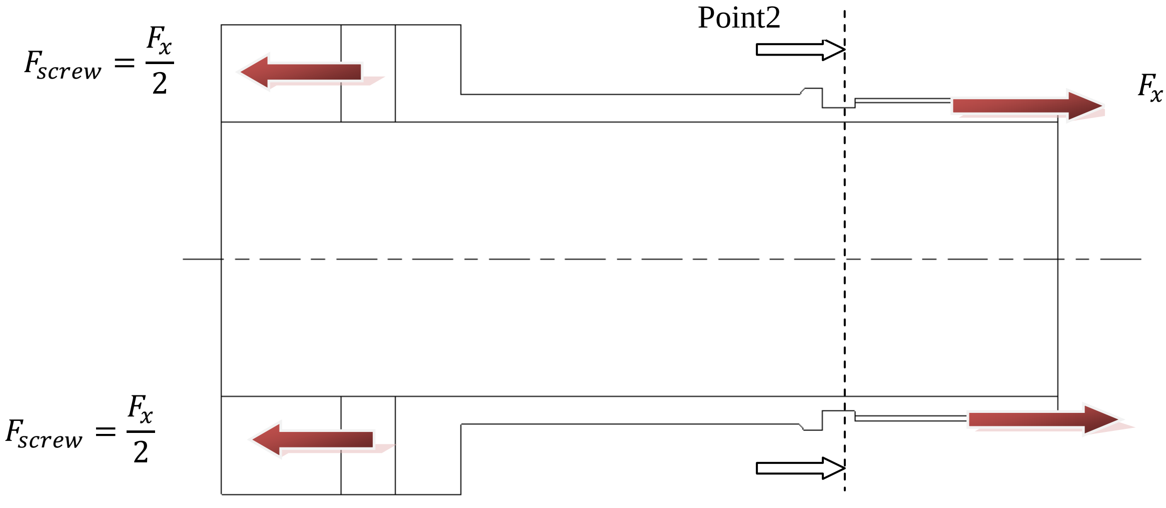


FIGURE 5 Forces in Part 6

5.7.1 **Weakest Point**

The weakest point in Part 6 can be located at the undercut before the thread (Point 2). Only the normal force F_x has to be transferred to the feed wheel (Part 1)

$$F_x = 3452.2N$$

The cross section that has to bear the force is an annulus with the inner diameter $d_{i2} = 44\ mm$ and an outer diameter of $d_{o2} = 48mm$.

$$A_{x2} = \frac{\pi * ({d_{o2}}^2 - {d_{i2}}^2)}{4} = \frac{\pi * (48\ mm^2 - 44\ mm^2)}{4} = 289.03\ mm^2$$

$$\sigma_{x3} = \frac{F_x}{A_{x2}} = \frac{3452.2\ N}{289.03\ mm^2} = 11.94\ N/mm^2$$

In table 4 section 5.4 of this thesis, the maximum allowable stress of Part 6 is listed

$$f_{allowable} = 236\ N/mm^2$$

$$11.94\ N/mm^2 \ll 236N/mm^2$$

Actually, the notch effects of the weakest point in Part 6 have to be taken into account as well, but as the occurring stress is very low and not even close to the maximum allowable stress, the part can still be seen as safe.

5.7.2 **Thread M52x2**

Due to the fact that the thread in Part 6 is a motion screw thread, the decisive parameter for the strength calculation is the surface pressure at the thread flanks. The M52x2 thread dimensions are:

$$H_1 = 0.54127 * P = 0.54127 * 2 = 1.0825\ mm$$
$$d_2 = 50.701\ mm$$

It is estimated that half of the thread length is in use and carries the load. The maximum allowable flank pressure can be determined as the same as in Part 6.

$$l_1 = 10\ mm$$
$$F_x = 3452.2\ N$$

$$p_{max} = \frac{12.5\ N/mm^2}{1.5} = 8.3\ N/mm^2$$

$$p_{flank} = \frac{F * P}{l_1 * \pi * d_2 * H_1}$$

$$p_{flank} = \frac{3452.2\ N * 2\ mm}{10\ mm * \pi * 50.701\ mm * 1.0825\ mm} = 4\ N/mm^2$$

$$4\ N/mm^2 \leq 8.3\ N/mm^2$$

5.8 Part 1, Feed Wheel

The feed wheel has to carry the load coming retaining ring, whereby the M52x2 thread annuls it by delivering it to the Part 6.

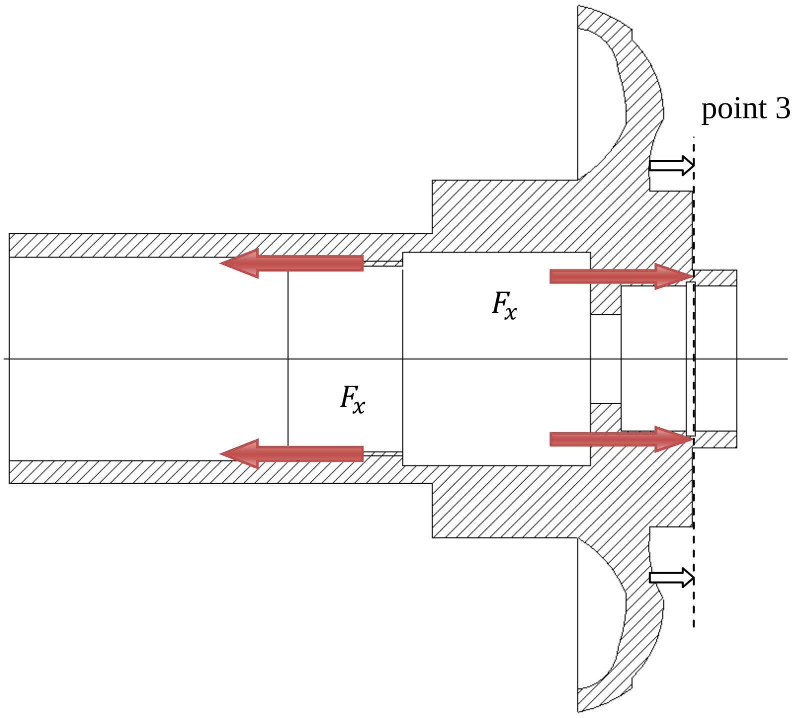


FIGURE 6 Forces in Part 1

5.8.1 **Weakest Point**

Weakest point of Part 1 is located directly behind the slot for the retaining ring (point 3). The force in x-direction has to be carried by the cross section with an inner diameter $d_{i3} = 31.5\,mm$ and an outer diameter of $d_{o3} = 38\,mm$.

The maximum allowable stress is taken from table 4 in section 5.4.

$$f_{allowable} = 236\,N/mm^2$$

$$A_{x3} = \frac{\pi * ({d_{o3}}^2 - {d_{i3}}^2)}{4} = \frac{\pi * (38\,mm^2 - 31.5\,mm^2)}{4} = 354.8\,mm^2$$

$$\sigma_{x3} = \frac{3452.2N}{354.8\,mm^2} = 9.73\,N/mm^2$$

$$9.73\,N/mm^2 \ll 236\,N/mm^2$$

As in Part 6, a notch effect occurs at the weakest point of Part 1. It can be neglected due to the low stress and the big reserve till the occurring stress reaches the maximum stress.

5.9 Standard Parts

5.9.1 Retaining Ring

The retaining ring has to be proven by comparing the existing force with the maximum allowable load for retaining rings. Usually, the slot is weaker than the retaining ring itself and thus, the slot value is decisive. The whole force (F_x) coming from the axial bearing has to be carried by the retaining ring.
The maximum allowable load for a DIN 472 retaining ring slot with the diameter of $d = 30\ mm$ can be compared with the table value (DIN 472 for d_1=30mm)

$$F_{smax} = \frac{11.3\ KN}{SF} = \frac{11.3KN}{1.5} = 7{,}53\ KN$$

$$3.452\ KN \leq 7{,}53\ KN$$

The maximum occurring force is lower than the maximum allowable force and thus, the retaining ring slot is safe.

5.9.2 Axial Bearing

The axial bearing SKF 51103 with the diameter of $d = 30\ mm$ has to be compared to the maximum allowable force as well. Equivalent to the value for the retaining ring, the load can be taken from a table (Roloff/Matek Tabellenbuch, page 144), whereby the value for static loads can be chosen. Because device is only in use for a small period of time, no big dynamic loads will appear. An indicated force from the shaft in x-direction (F_x) is acting on the axial bearing.

$$C_0 = 19.6KN$$

$$C_{max} = \frac{19.6KN}{SF} = \frac{19.6KN}{1.5} = 13.06KN$$

$$3.452KN \leq 13.06KN$$

The part can be seen as safe too since the maximum allowable force is higher.

6 EXAMINATION WITH ANSYS® WORKBENCH

6.1 About the Use of FEM Methods and ANSYS® Workbench

Several methods can be used for the dimensioning of parts according to the Pressure Equipment Directive annex I section 2, including analysis methods. The computer based Finite Element Method (FEM) is an analysis method which is nowadays very often applied in several work environments of mechanical engineering. With the help of FEM, it is possible to find acceptable solutions for complex elastic-mechanic coherences inside a part or in assemblies that cannot be solved with classical mechanic methods.

FEM is basically working with a finite amount of elements, which are connected together while forming different element types (3d solid element model, shell model…) to create a mesh. Since elements have a certain shape it is not possible to create a completely realistic model of a part, whereas the accuracy increases with the element amount. By creating approaching functions between 2 knots and by stetting of certain initial and boundary conditions, matrixes are formed and strains of the knots are described. From the occurring strains, forces and stresses can be calculated. Especially in more elaborated models, the calculation time needs to be taken into account while defining element types and approaching functions.

As already mentioned, the method of FEM contains inaccuracies and the results have to be checked very precisely. Especially the contact areas, the initial conditions, as well as the too rough mesh are very often creating unrealistic values which disable the results.

ANSYS® Workbench is an application that uses the method of FEM for calculating strains and stresses. It has a clearly arranged desktop workplace and is thus comparatively easy to handle while results can be visualized graphically.

In this thesis, the student version of ANSYS® Workbench is used, which is confined regarding its properties and accomplishments. The creations of assemblies as well as fine meshes are limited to a certain extent.

6.2 Adjustments in ANSYS® Workbench

Before the modelling and the simulation can start, the adjustments of ANSYS® Workbench should be checked and evaluated. Besides the standard material values, such as the module of elasticity or transversal contraction, especially the mesh type needs to be observed.

In the following simulations, 3-D 10-Node Tetrahedral Structural Solid Elements (SOLID187), which are tetrahedral 3d solid elements with 3 degrees of freedom and quadratic displacement behaviour, have been used. Generally, the convergence to the exact solution of an FEM simulation depends on the approaching function, the amount of elements and the element type, whereby quadratic approaching functions are usually more precise than linear functions.

(http://www.sharcnet.ca) (Grundlagen und Anwendungen…, page 170-174)

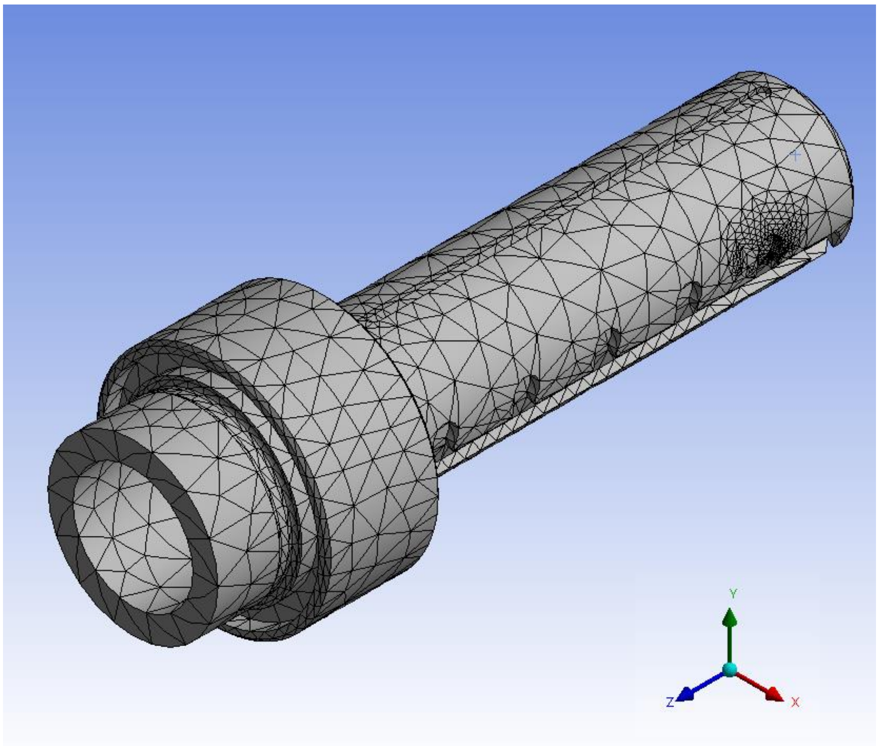
6.3 Examination of Part 10, Body

As already mentioned in the strength calculations (section 5), these investigations have to be seen as examinations and the company TONISCO System Oy has to decide in which way they are going to use them. In general, FEM examinations are not required from the VdTÜV and for the EC Type-Examination approval, but as the use of simulation programs is very common nowadays, it is reasonable to analyse a part by an application like ANSYS®.

Part 10 is the most complex part in the device and especially the contact area between the power screws and Part 10 are recommended to be observed with FEM. Since a high force is acting on a small area it is considered highly stressed.

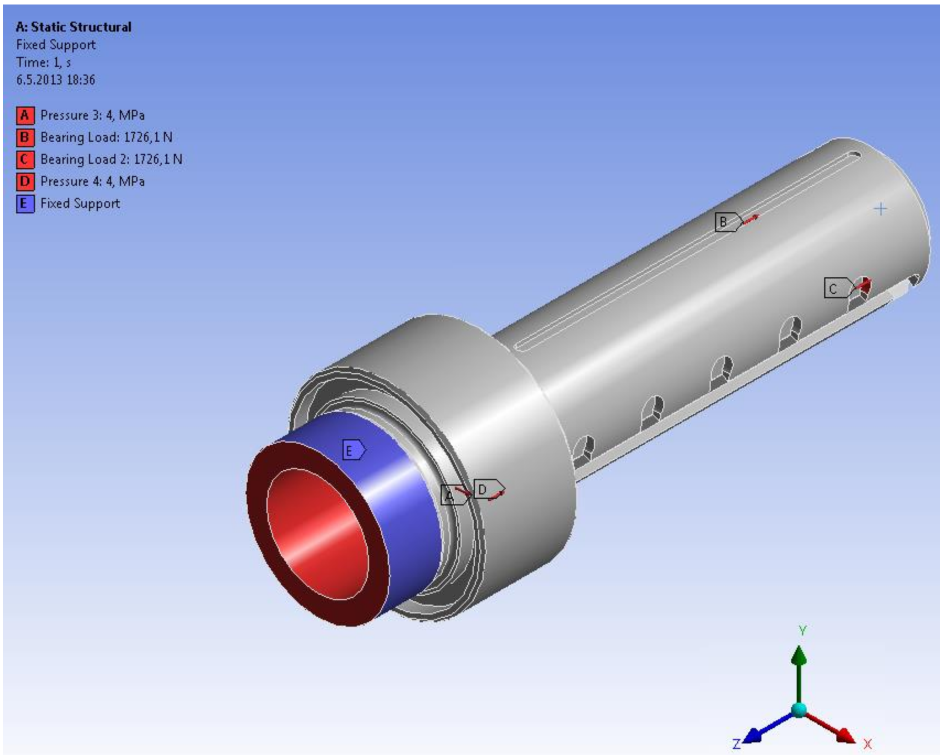
6.3.1 **Modelling of Part 10**

After creating a 3D solid model of Part 10 with the correct settings, a mesh has to be formed. This mesh should be as fine as possible while taken the calculation time into account. Because the contact section of the screw forces are highly stresses, it is reasonable to add refineries as visible in picture 7.



PICTURE 7 Mesh with refineries of Part 10

After producing a mesh, certain forces and initial conditions have to be determined whereby a realistic scenario that describes the reality as accurate as possible, has to be created.

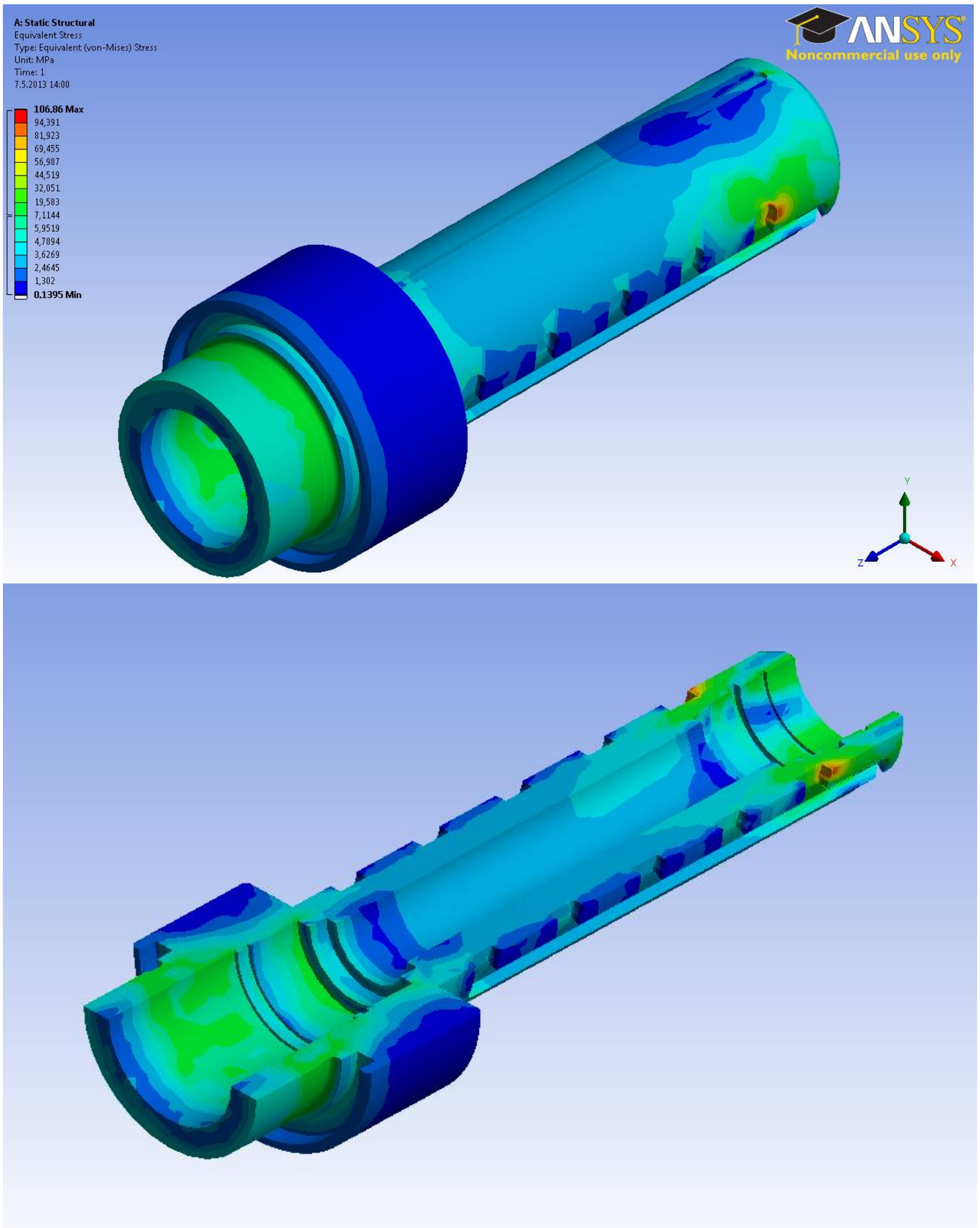


PICTURE 8 Forces and initial conditions in Part 10

The contact areas with the screws (C, D) can be modelled as bearing loads and the thread as a fixed support (E) as seen in picture 8.

6.3.2 Stress Results in Part 10

With the defined loads and initial conditions, a solution can be calculated. As different states of stresses are interfering, it is reasonable to visualize the equivalent stress hypothesis of von Mises for ductile materials (shape modification hypothesis).



PICTURE 9 Equivalent Stress in Part 10

As shown in picture 9, the maximum stress occurs at the location of force application of the screws. The maximum equivalent stress is $\sigma_v = 106.86\, N/mm^2$. This value can be compared with the maximum allowable stress shown in table 4section5.4.

$$106.86\ N/mm^2 \leq 426\, N/mm^2$$

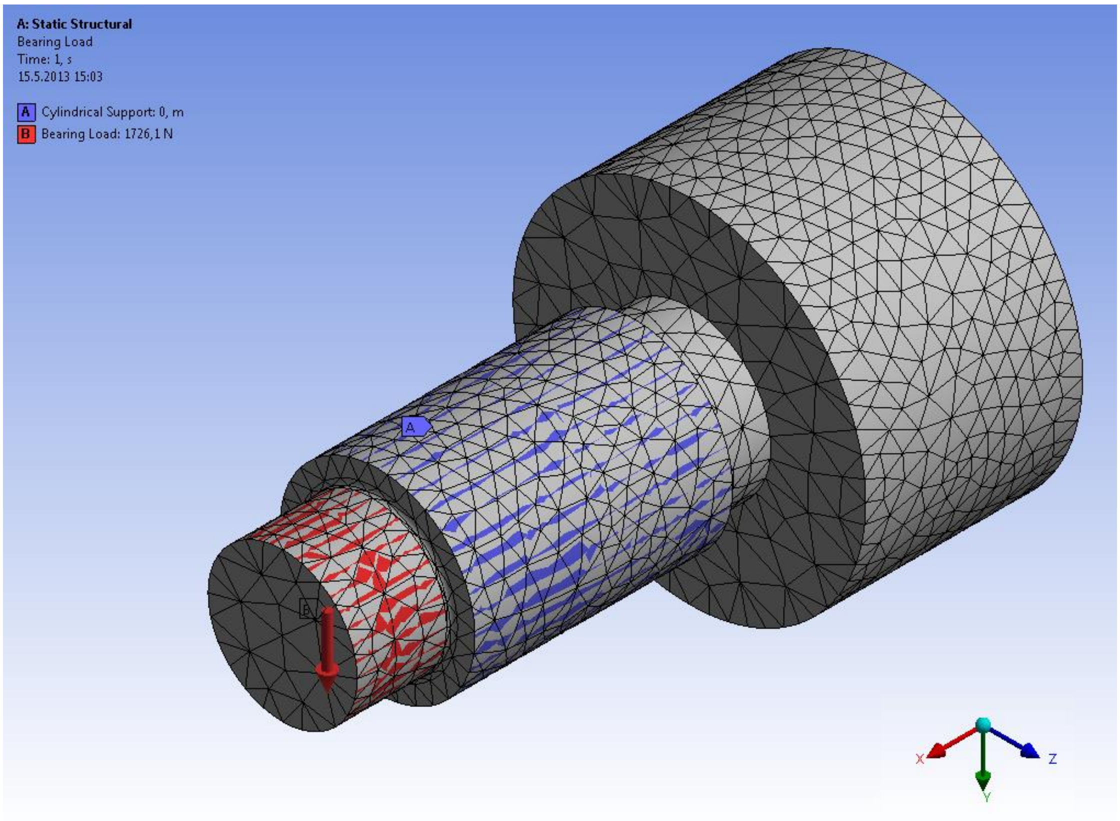
Even though the occurring stress is lower than the maximum allowable stress, the correctness of the results has to be observed very detailed. In this model, the assumption of the screw forces as bearing loads, as well as the assumption of the thread as a fixed support can be possible inaccuracies. But even though the result is not completely reflecting the reality, the part can be seen as safe, as there is still a lot of reserve between the maximum stress and the maximum allowable stress.

6.4 Examination of Part 9, Power Screws

The calculations according to basic formulas have shown very high stress peaks visible in section 5.6. Because of that, the part is highly recommended to be more observed and the results shall be compared. Changing the radius of the notch just a bit increases or decreases the value of the maximum stress dramatically and thus, it is reasonable to examine several different radii. The results should also help to choose the correct forming tool. Since most of the usual forming tools have at least a really small radius, a completely sharp corner is not realistic.

6.4.1 **Modeling and Visualization with ANSYS®**

Like in the previous part (section 6.3.1), the 3d solid model has to be drawn and a mesh has to be created. As initial and loading conditions, a cylindrical support is added while a bearing load with a force of$F_x/2 = 1726.1N$acts on the screw. As visible in picture 10, the initiated force $F_x/2$acts as a bearing load in y-direction. Since the M8 thread can't be modelled exactly, the pitch diameter is inserted instead.

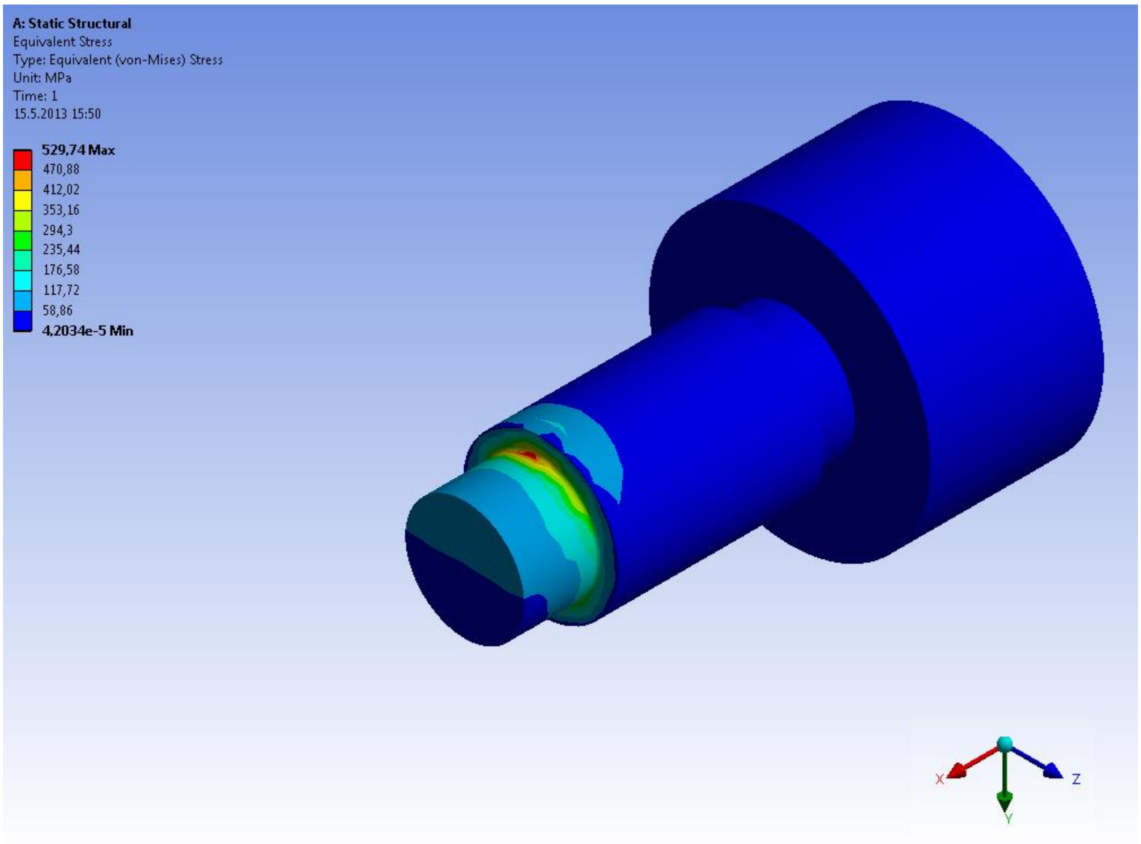


PICTURE 10 Forces, initial conditions and a created mesh of Part 9

Since several different values need to be calculated for different notch radii, it should be defined as a parameter. As especially the maximum of the equivalent stress is the value of interest, it should be displayed.

6.4.2 **Stress Results in Part 9**

The equivalent stress (von Mises) can be visualized graphically in the simulation application of ANSYS® Workbench as it is shown in picture 11.



PICTURE 11 Equivalent stress in Part 9 and a notch radius of $r_{notch} = 0{,}2mm$

The highest stress is located at the perimeter in the area of the notch. For a notch radius of$r_{notch} = 0.2mm$, ANSYS® Workbench shows even higher results. As a reason for the differences, a too rough mesh or interference of different state of stresses, which haven't been taken into account in the hand calculations, are assumed. Generally, a radius of $r_{notch} = 0.2mm$ can be seen as sufficient to carry the assumed load from the inner pressure and the drilling. Nevertheless, for a more precise calculation with ANSYS®, non-linear FEM needs to be used and the mesh needs to be finer, whereby other element types can be selected.

$$529.74\ N/mm^2 \leq 648N/mm^2$$

6.4.3 **Examination of the Notch Radius**

The equivalent stress varies significantly with the size of the notch radius which, in turn, depends on the radius of the forming tool. In general, a smaller radius causes a higher stress in the part, whereby a very small radius seems to be an exception. To find a minimum allowable radius, FEM can be used. Therefore the radius and the maximum equivalent stress have to be defined as parameters, which are displayed in a table. Several notch radii have been observed at different stages, from $r_{notch} = 0{,}01mm$ to $r_{notch} = 0{,}6mm$.

Table of Design Points

	A	B	C	D	E
1	Name	P4 - NotchRadius.FD1	P3 - Equivalent Stress Maximum	Exported	Note
2	Units		Pa		
3	Current	0,2	5,2974E+08		
4	DP 1	0,01	6,1602E+08		
5	DP 2	0,1	6,9514E+08		
6	DP 3	0,2	5,2974E+08		
7	DP 4	0,3	4,4514E+08		
8	DP 5	0,4	4,2365E+08		
9	DP 6	0,5	3,9956E+08		
10	DP 7	0,6	3,4898E+08		
*					

TABLE 5 Equivalent stresses for different notch radii

From table 5 it can be seen that the notch radius needs to be 0,2mm to be lower than the maximum allowable stress with plastic deformation and to be on the safe side.

With $$r_{notch} = 0.2\ mm \Rightarrow\!\!> \sigma_v = 529.7\ N/mm^2 \leq 648\ N/mm^2$$

Against the assumed theory, a very small radius of $r_{notch} = 0.01\ mm$ doesn't create the highest stress, but since an exact examination would exceed the scope of this thesis and it is hardly possible to create such small radii in reality, it will not be observed at this point.

7 STRESS RESULTS AND SECURITY VALUES

For a better overview it is very useful to list the stress and security values of the parts and connections in a table. The security value is calculated by dividing the maximum allowable stress/force/pressure by maximum occurring stress/force/pressure.

TABLE 6 Stress results and security values

Part or connection	**Maximum allowable value**	**Maximum occurring value**	**Safety value**
Thread in Part 10	$8.3\ N/mm^2$	$7.17\ N/mm^2$	1,157
Weakest point in Part 10	$426\ N/mm^2$	$16.89\ N/mm^2$	25,22
Force screws, Part 9			
Shearing	$346.41\ N/mm^2$	$96.86\ N/mm^2$	3,576
Stress in notch	$648\ N/mm^2$	$443.945\ N/mm^2$	1,46
Hole bearing	$152.86\ N/mm^2$	$15,41\ N/mm^2$	9,92
Thread in Part 6 and 1	$8.3\ N/mm^2$	$4\ N/mm^2$	2,075
Weakest point in Part 6	$236\ N/mm^2$	$11.94\ N/mm^2$	19,765
Retaining ring in Part 1	$7{,}53\ KN$	$3.452\ KN$	2,183
Bearing in Part 1	$13.06\ KN$	$3.452\ KN$	3,783
Weakest point in Part 1	$236\ N/mm^2$	$9.73\ N/mm^2$	24,25

Since every safety value is ≥ 1 , all parts can be seen as safe.

8 DISCUSSION

The purpose of this thesis in the first place has been to find the responsible authority for EC Type-Examination approvals, to create a contact and to figure out the requirements for the product to be certificated. In the second step, the requirements needed to be fulfilled and documents to be provided. During this process, it was necessary to familiarize with the regulations from the PED and the notified body, the VdTÜV. Moreover, a calculation of the pressure part and an FEM examination needed to be done since the existing documents regarding strength calculations are quite old.

The work for this thesis showed that the whole EC-Type Examination approval is an elaborated process and the original aim of getting it within this limited period of time couldn’t be achieved. Especially, since the whole procedure has to be done according to the VdTÜV and PED regulations, it was a big effort to observe the validity of the old documents and possibly correct and update them. Furthermore, it was necessary produce certain documents that were needed.

Regarding the calculation and the simulation part of the thesis work, it can be said that the TONISCO Jr. is surely a safe product as the occurring stresses never exceeded the maximum allowable stresses. Even though some values needed to be estimated, the load assumptions that have been made are all the time on the safe side and as much standards as possible have been used.

As a next step, the whole process should be completed to get the approval form the VdTÜV. Therefore, it is necessary to create or update the required documents and submit them to the contact person of the notified body. In the beginning, the certificate for the TONISCO Jr. should have the priority, but if the submitted documents have been accepted, TONISCO System Oy should thing about a certificate for their other products by using a similar way. As mentioned in this thesis, the focus in there lays on the approval for district heating. However, after all TONISCO drilling devices got the certificate for that field of use, the company may think about a certification for other work environments, such as gas distribution.

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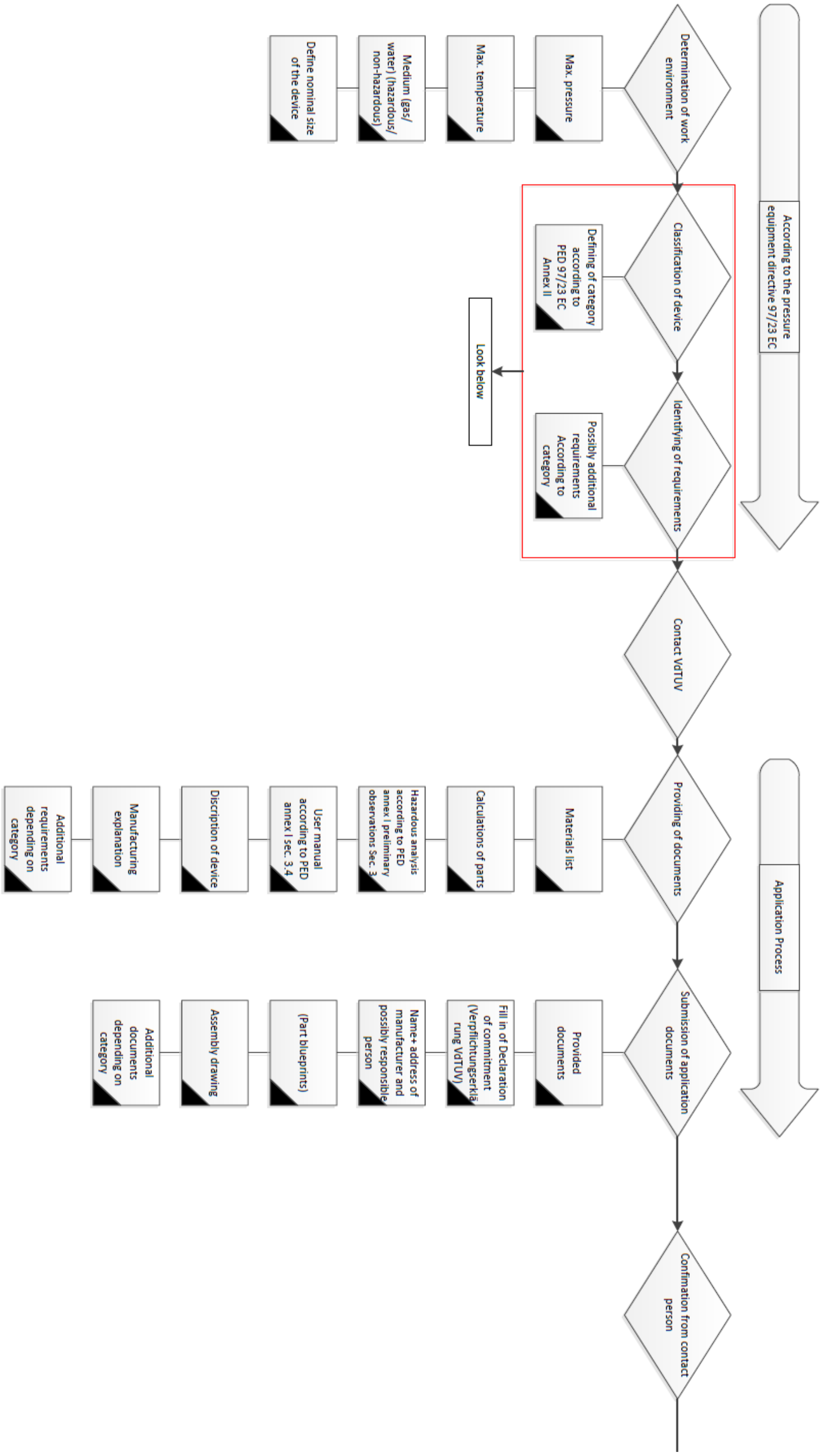
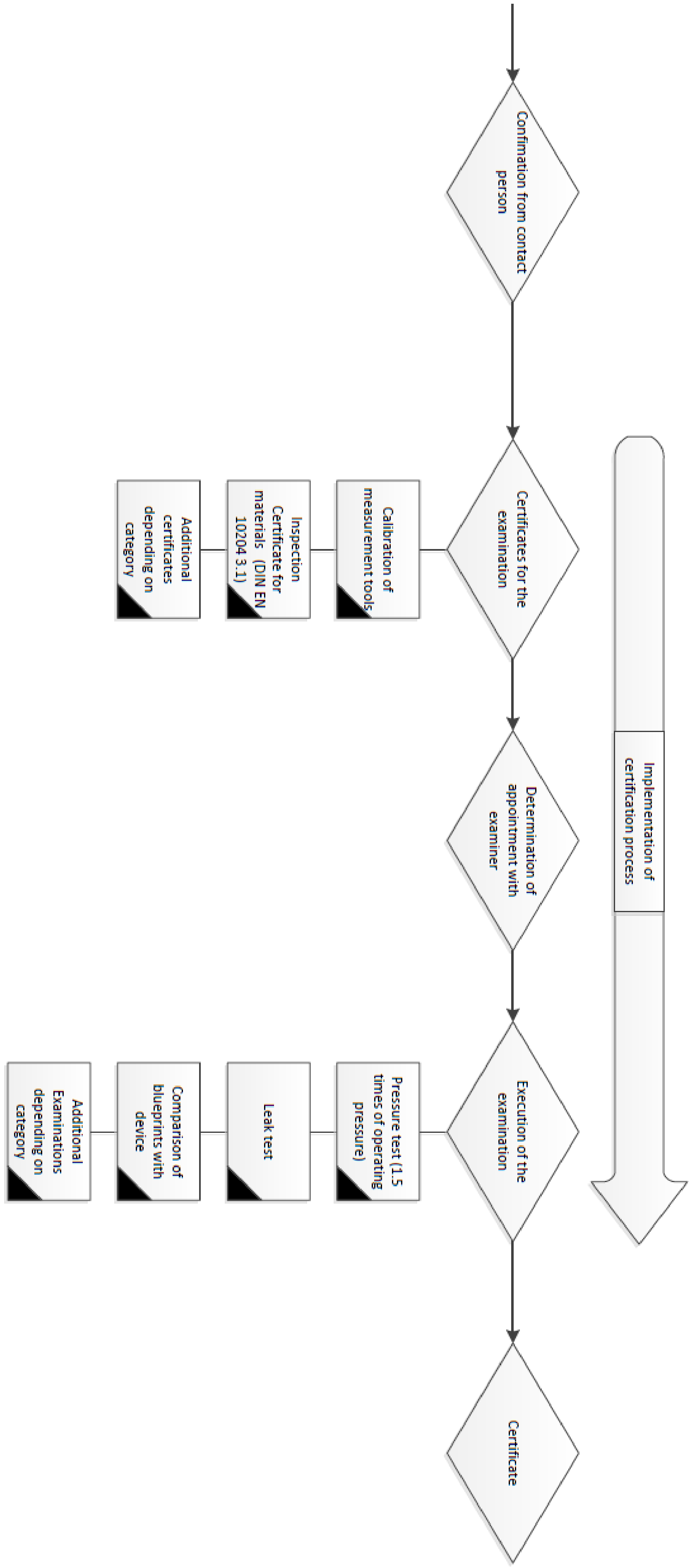
10 ACKNOWLEGEMENT

I would like to thank Harri Laaksonen for the supervising and the very helpful feedback but also the company TONISCO System Oy for offering this project and the help to get known of their products and their function. Furthermore, I want to express gratitude to the TAMK and the members that made it possible for me to work on this project, but also the contact persons from the VdTÜV that gave the important information about the certification process.

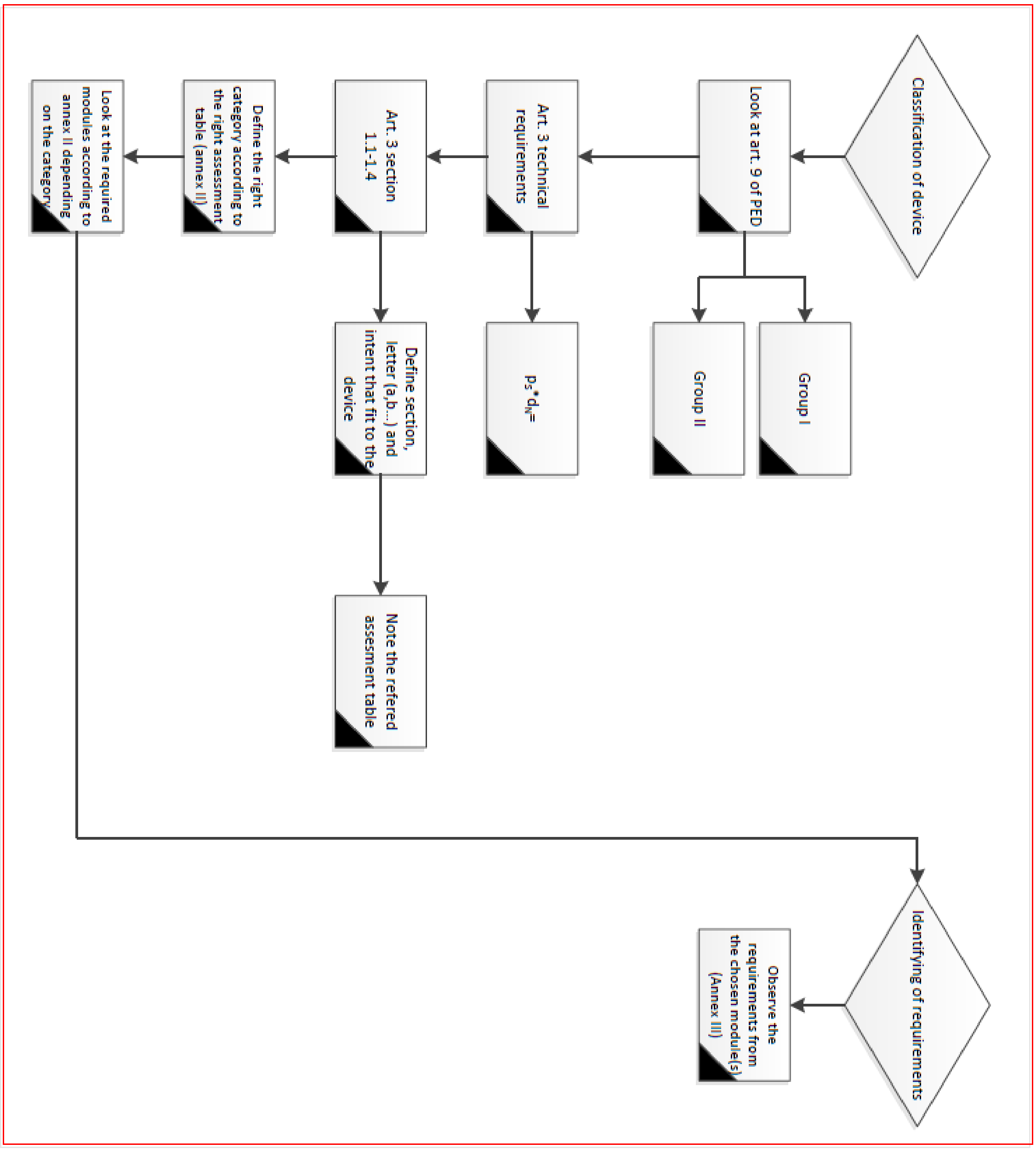
APPENDICES

Appendix 1. Project Plan





Confirmation from contact person
Certificates for the examination
Calibration of measurement tools
Inspection Certificate for materials (DIN EN 10204 3.1)
Additional certificates depending on category
Determination of appointment with examiner
Implementation of certification process
Execution of the examination
Pressure test (1.5 times of operating pressure)
Leak test
Comparison of blueprints with device
Additional Examinations depending on category
Certificate


Classification of device
Look at art. 9 of PED
Group I
Group II
Art. 3 technical requirements
P_S*d_N=
Art. 3 section 1.1-1.4
Define section, letter (a,b…) and intent that fit to the device
Note the refered assesment table
Define the right category according to the right assessment table (annex II)
Look at the required modules according to annex II depending on the category
Identifying of requirements
Observe the requirements from the chosen module(s) (Annex III)

Appendix 2. Extract of the Translated Manual

"…Herstellen von Rohrabzweigungen mit der TONISCO Jr. Anbohrarmatur

1. Das Ende der TONISCO Sperre ist an die Form des Hauptrohres durch spanende Bearbeitung anzupassen und elektrisch anzuschweißen. Hierbei ist auf die korrekte Position und Richtung zu achten. Die Anbohrarmatur kann im Anschluss an der Sperre angebracht werden.
2. Die Festig- und Dichtigkeit der gesamten Anbohrarmatur wird mittels Drückprüfung getestet. Ist diese erfolgreich, wird die Bohrung durchgeführt. Der Bohrvorgang beginnt mit einer Zentrierbohrung und einer anschließenden Bohrung mit Lochsäge.
3. Nach Abschluss des Bohrvorgangs sollte die Spindel zusammen mit der aus dem Hauptrohr abgetrennten Lochscheibe aus dem Schließbereich der Sperre entfernt werden.
4. Das Ventil wird durch die Sperrscheibe abgedichtet, woraufhin die Bohrarmatur demontiert werden kann.
5. Die zuvor angepasste Rohrabzweigung kann nun an die Sperre angeschweißt und die Festig- und Dichtigkeit aller Schweißnähte durch eine Druckprobe gegen die Sperrscheibe überprüft werden.
6. Im letzten Schritt wird die Sperrscheibe entfernt und die Sperrscheibennut verschweißt. Gummidichtungen im Inneren der TONISCO Sperre verhindern eine Leckage im Bereich der Sperrscheibe.

Ein passendes Bohrfutter (Pos. 30 oder 31) sollte ausgewählt und die Lochsäge daran angeschraubt werden. Wenn eine Lochsäge > 30mm verwendet wird, müssen die Sicherheitsstifte des Bohrfutters durch die Löcher am Boden der Lochsäge geführt werden. Das Bohrfutter kann nun an die Spindel (Pos.36) angeschraubt werden.

Der Zentrierbohrer wird durch das Loch im Bohrfutter geführt. Die Sicherungsschraube(Pos. 34) wird an der Stelle der Zentriernut in der Bohrstange mit einem 4mm Sechskantschlüssel (Pos. 23) angeschraubt. Nach dem Anschrauben sind die Harken des Zentrierbohrers zum Auffangen der Lochscheibe zu untersuchen. Die Harken müssen sich ausspreizen lassen, damit die ausgeschnittene Lochscheibe aufgefangen werden kann.

Die Zähne der Lochsäge und die Spitze des Zentrierbohrers sollten mit einer dünnen Schicht „TONISCO Cutting Paste" bestrichen und die Spindel mit „TONISCO Sealant Lubricant" eingefettet werden.

7.

Die Vorschubhülse (Pos.1) ist von der Bohrvorrichtung zu entfernen und die Dichtung in der Hülse zu überprüfen und einzufetten. Falls diese beschädigt sein sollte, muss sie ersetzt werden.

Der für das Ventil passende Adapter wird ausgewählt und seine Dichtungen untersucht und eingefettet. Zum Schluss wird die Spindel mit Bohrfutter, Lochsäge und Zentrierbohrer, wie in der Zusammenbauzeichnung dargestellt, eingesetzt.

8. Installation der Anbohrarmatur

Der Spindel mit Bohrfutter, Lochsäge und Zentrierbohrer kann nun in die Bohrkammer des Gestells (Pos.10) geschoben werden. Der passende Adapter wird auf die TONISCO Sperre aufgeschraubt und die Anbohrarmatur mit dem Adapter verbunden.

Nach der Montage sollten Sie erneut überprüfen, dass der Innendurchmesser des Ventils tatsächlich groß genug für die ausgewählte Lochsäge ist. Hierzu sollten Sie die Spindel vor- und zurückschieben, bis der Zentrierbohrer das Hauptrohr berührt. Beim Zurückziehen der Spindel ist es ebenfalls möglich sich zu vergewissern, dass sich der Zentrierbohrer nicht zwischen dem Schließmechanismus des Ventils befindet, indem Sie die Sperrscheibe vorsichtig einschieben und gleichzeitig überprüfen, dass sich nichts im Zwischenraum aufhält.

9.

Die sich auf der Einstellhülse(Pos. 6) befindende Vorschubhülse wird komplett zurückgeschraubt. Die Vorschubeinheit sollte an ihrem definierten Platz angebracht werden, so dass sich die Sicherungsstifte der Gleitschiene in den Führungsnuten des Gestells bewegen können. Die Vorschubeinheit wird so lang auf das Gestell in Richtung des Hauptrohres geschoben bis der Wellenabsatz der Spindel das Axiallager berührt. Die Vorschubhülse sollte so weit aufgeschraubt werden, bis die Einstellhülse in der nächsten Nut verankert werden kann. Falls nur die letzte oder vorletzte Nut als Verankerung verfügbar ist, sollte die Spindelverlängerung(Pos. 37 oder 38) genutzt und die Vorschubhülse erneut montiert werden.Zuletzt muss sichergestellt werden, dass Kontrollhahn(Pos. 19) am Manometer(Pos. 22) geschlossen sind.

10.

Bringen Sie die Antriebseinheit am oberen Ende der Spindel an und verschrauben Sie das Getriebe(Pos. 49) mit der Spindel durch eine 8mm Schraube (Pos. 29). Stellen Sie für die Zentrierbohrung die höchste Geschwindigkeit am Bohrgerät ein.

11. Verzweigen mit der „TONISCO Jr.“

Der Bohrvorgang beginnt mit einer Zentrierbohrung, wobei der Vorschub sehr vorsichtig erfolgen muss.Um eine ordentliche Zentrierung zu gewährleisten, sollte der Vorschub am Anfang sehr langsam erfolgen. Bei manuellem Vorschub ist das Risiko den Zentrierbohrer zu beschädigen sehr groß, da mit geringer Zerspanungsgeschwindigkeit gearbeitet wird. Falls der Bohrer trotz aller Mühe nicht weiter schneidet, kann der Bohradapter demontiert und der Zentrierbohrer inspiziert werden. Es ist jedoch sehr wichtig zuvor darauf zu achten, dass der Zentrierbohrer nicht bereits durch das Rohr durchgebohrt hat. Für diese Überprüfung kann der Kontrollhahn vorsichtig geöffnet werden.
Im Allgemeinen ist es möglich den Kontrollhahn kann während des Bohrens geöffnet zu lassen…“